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DYNAMIC TEST OF GEARS MANUFACTURED BY ADVANCED FORGING
TECHNIQUES

UNITED AIRCRAFT CORP.

PREPARED FOR
ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY

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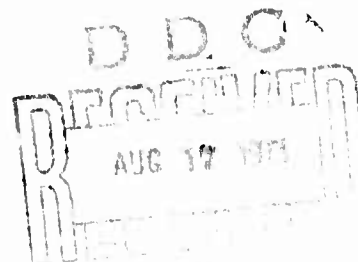
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USAAMRDL TECHNICAL REPORT 73-13

DYNAMIC TEST OF GEARS MANUFACTURED BY ADVANCED FORGING TECHNIQUES

By
Harold K. Frint

May 1973



EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-69-C-0060
SIKORSKY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
STRATFORD, CONNECTICUT

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This report was prepared by Sikorsky Aircraft, Division of United Aircraft Corporation, under the terms of Contract DAAJ02-69-C-0060. The technical monitors for this contract were Messrs. James Gomez and L. M. Bartone, Technology Applications Division.

The objective of this program was to conduct a preliminary evaluation of the dynamic fatigue strength of spur gear teeth produced by advanced forging techniques.

A laboratory evaluation of a limited number of gear samples using three advanced gear forging processes and a conventional (pancake) forging process was conducted. Results indicated that the best advanced forging process shows only a small advantage over the conventional forgings. The results and conclusions contained herein do not represent a thorough evaluation of advanced forged gears, due to the limited number of samples that were tested.

Task 1G162207AA7201
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Final Report
Sikorsky Engineering Report 50680

By
Harold K. Frint

Prepared by
Sikorsky Aircraft
Division of United Aircraft Corporation
Stratford, Connecticut

for
EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

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SUMMARY

The results of a 38-month gear fatigue research program are reported herein. The purpose of this research effort was to evaluate the dynamic fatigue strength of spur gear teeth produced by advanced forging techniques in comparison to those manufactured by conventional forging methods.

This program compliments a previous program conducted by Sikorsky Aircraft which evaluated the single-tooth static and fatigue strengths of spur gears produced by these same forging methods. The results of that program are contained in USAVLABS Technical Report 69-11 (Reference 2).

All of the advanced forging processes evaluated in this program used high-energy-rate forging techniques which produced gear blanks with integrally forged teeth. The die design and type of press were different for the three advanced processes; however, they produced similar as-forged gear blanks except for some variations in flash formation.

On the basis of the results obtained, the best advanced gear forging process shows an advantage over the conventional forging process which is in the order of 5 to 8 percent.

FOREWORD

This report covers a comparative evaluation of spur gears manufactured from conventional and high-energy forging techniques. The high-energy forgings produced gear blanks with integrally forged teeth, whereas the conventionally produced gears were hobbled from pancake-type forgings. The evaluation involved dynamic fatigue tests in a closed-loop regenerative test rig. The program was conducted during the 38-month period from April 22, 1969 to June 22, 1972 for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USAAMRDL) under Contract DAAJ02-69-C-0060. This program is a continuation of a previous single-tooth fatigue test program conducted on spur gears manufactured from the same forging dies for USAAMRDL under Contract DAAJ02-67-C-0014. The work was authorized by DA Task 1G162207AA7201.

The program was conducted at Sikorsky Aircraft, Stratford, Connecticut. Principal investigators for the program were Lester R. Burroughs, Supervisor, Transmission Design and Development Section, Harold K. Frint, Senior Analytical Engineer, and J. Lucas and J. Rayno of the Materials Section.

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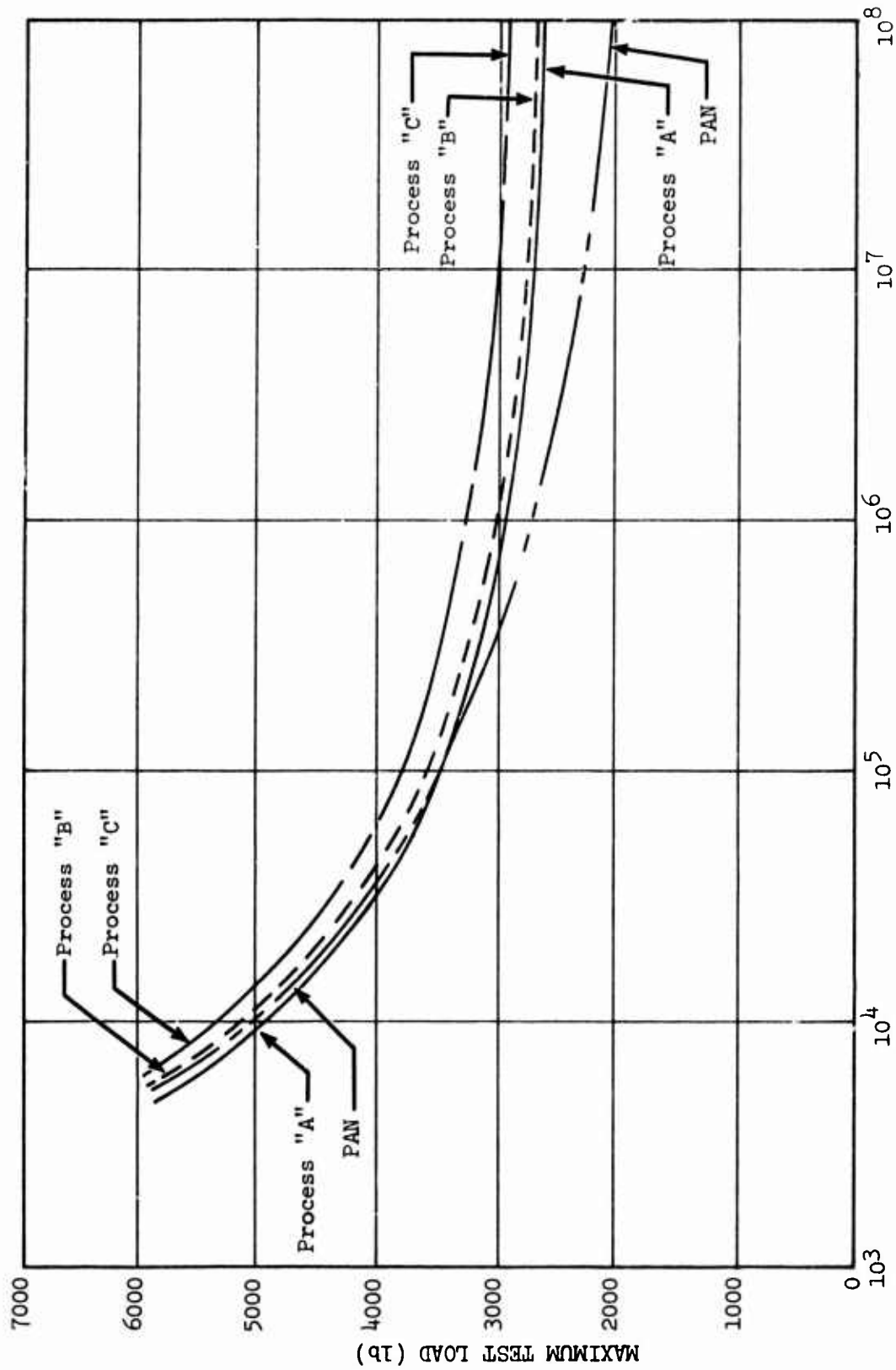
LIST OF SYMBOLS

d	deviation of test point from the mean curve
E	endurance limit at an infinite number of cycles, psi or lb
gpm	gallons per minute
N	number of cycles
n	number of test points
R_c	Rockwell hardness number
S	stress, psi, or tangential tooth load, lb
S/N	serial number
s	unbiased standard deviation, lb
TIF	lowest position of true involute form
\bar{X}	mean fatigue strength, lb
S/\bar{X}	coefficient of variation
β	material constant in S-N relation
γ	material constant in S-N relation
\bigcirc	test point, heat 1
$\bigcirc \rightarrow$	nonfailure point
Δ	test point, heat 2

INTRODUCTION

Along with the trend in recent years toward higher power-to-weight ratio aircraft engines, transmission designers are constantly being challenged to improve the power-to-weight ratio of drive train components, thereby increasing the power train contribution to aircraft effectiveness (increased payload) for the same reliability. Similarly, power train reliability could be improved for the same effectiveness. This is particularly true in helicopter designs, where transmission weight is approximately 10 percent of aircraft weight. These requirements necessitate an increasing use of higher strength as well as lighter weight materials. Recent manufacturing and processing developments have introduced precision forging techniques that minimize initial machining operations, thus lowering material and manufacturing costs. In addition, higher ultimate tensile strengths and yield strengths result from better control of grain size and orientation (grain flow). Several programs to determine the comparative fatigue properties of case-hardened gears produced by conventional and high-energy forging techniques, which produce gear blanks with integrally forged teeth, have been conducted in recent years. On one of these programs (Reference 1), high energy rate forged (HERF) gears and gears cut and ground from either bar stock or simple upset forgings were tested at two stress levels. At both stress levels, the gears forged with integral teeth had average fatigue lives substantially greater than those of the machined gears. A similar result was obtained in a previous program conducted by Sikorsky Aircraft (Reference 2), in which single-tooth bending fatigue strengths of gear teeth manufactured by three advanced forging processes and one conventional forging process were determined and compared. Increases in single-tooth bending endurance limit were found to vary from 24 to 44 percent when comparing the integrally forged teeth with the conventionally produced gears. The results of these single-tooth tests are shown in Figure 1. "A", "B", and "C" refer to the three advanced forging processes and PAN refers to the conventional pancake forging.

This report presents the results of a dynamic test program, conducted by Sikorsky Aircraft as a continuation of the efforts summarized above, to evaluate the comparative fatigue strengths of gears produced by advanced forging techniques. The test gears designated as Heat 1 were manufactured from the same lot of conventional and advanced forgings tested in the Sikorsky single-tooth program of Reference 2. Twenty-four additional test gears, 12 from the conventional process and 12 from an advanced forging process, were designated as Heat 2 and tested to provide supplementary data to better define the S-N curve shapes.



CYCLES TO CRACK DETECTION

Figure 1. Results of Single-Tooth Fatigue Test.

FORGING MANUFACTURE

DISCUSSION

The goal of the previous single-tooth program (Reference 2) was to evaluate the comparative fatigue strengths of spur gears produced by three advanced (high-energy) forging processes. The resultant ranking of these advanced processes, and the conventional pancake forging, showed that all of the high-energy processes had endurance limits which were higher than that of the conventional forging (refer to Figure 1). It was understood, however, that the final proof of the superiority of the high-energy forging process over the conventional process would depend upon the results of a dynamic gear test program in which test conditions more closely approximated service conditions. With this in mind, a sufficient quantity of raw material was purchased, at the outset of the single-tooth program, to supply gear blanks for both a single-tooth and a dynamic test program (Heat 1). The material for Heat 2 was purchased separately at the conclusion of the initial tests.

RAW MATERIAL

The material selected was AMS 6265, an SAE 9310 vacuum-melt carburizing steel widely used for aircraft gearing. Material for both the single-tooth and dynamic test programs was received in 12-foot lengths with a certified copy of the mill test sheet indicating heat number, chemical analysis, and physical properties of the raw material. The material was inspected for conformity, and an ample quantity was shipped to each forging manufacturer for processing.

DIE DESIGN

Since it was anticipated that a dynamic test program would follow the original program, enough gear blanks were produced from the same dies for both programs. Gear blanks for the gears of Heat 2 were produced from the same dies as those of Heat 1.

To reduce the number of machining operations from initial forging to finished gear and to insure that a maximum of the beneficial surface grain flow remained after machining, it was considered desirable to design the forging dies so that only a minimum of stock removal was necessary to produce the finished gear. The more important areas for maintaining a uniform grain flow were the tooth flank and root fillet areas. On this basis, each forging manufacturer selected to participate in the single-tooth and dynamic test programs was instructed to produce gear blanks with a tooth profile within 0.010 to 0.014 inch of the finished tooth size as defined by Figure 2.

FORGINGS

Other than the requirements for grain flow, gear tooth dimensions, toler-

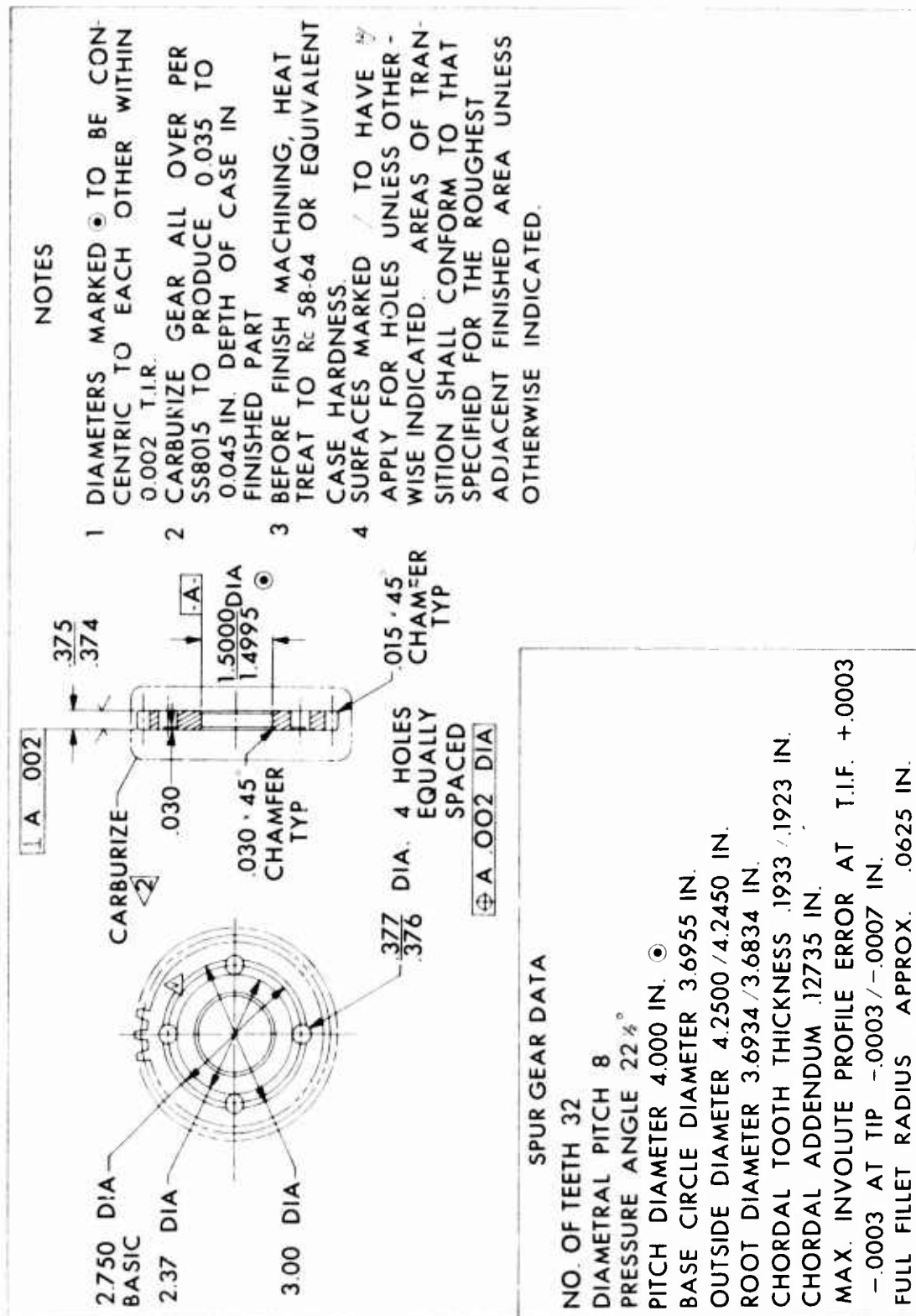


Figure 2. Test Gear, Dynamic Fatigue Test.

ances, and as-forged gear blank size, the forging procedures to be used by each manufacturer were established by the individual forging company to accommodate his own forging equipment and experience.

Pancake Forgings

The conventional pancake forgings were forged by a steam hammer in several successive blows at an initial temperature of 2200°F. After forming, the forgings were process-annealed at 1200°F for one hour. They were then air-cooled, producing a Brinell hardness of 197. The forging blank produced by this method is shown in Figure 3.

Advanced Forgings

Process "A"

The procedure used by this forging source consisted of an initial billet upset following an atmospheric heating to 1850°F. After air-cooling, the billet was grit blasted to remove scale and decarburization. The final forging operation was preceded by a heat soak at 1750°F. The gear blank produced by this process is shown in Figure 4.

Process "B"

Of the three advanced forging processes evaluated, only this source obtained a final gear blank with one forging hit. This forging operation was preceded by heating the billets in an inert atmosphere to 2100°F. This gear blank is shown in Figure 5.

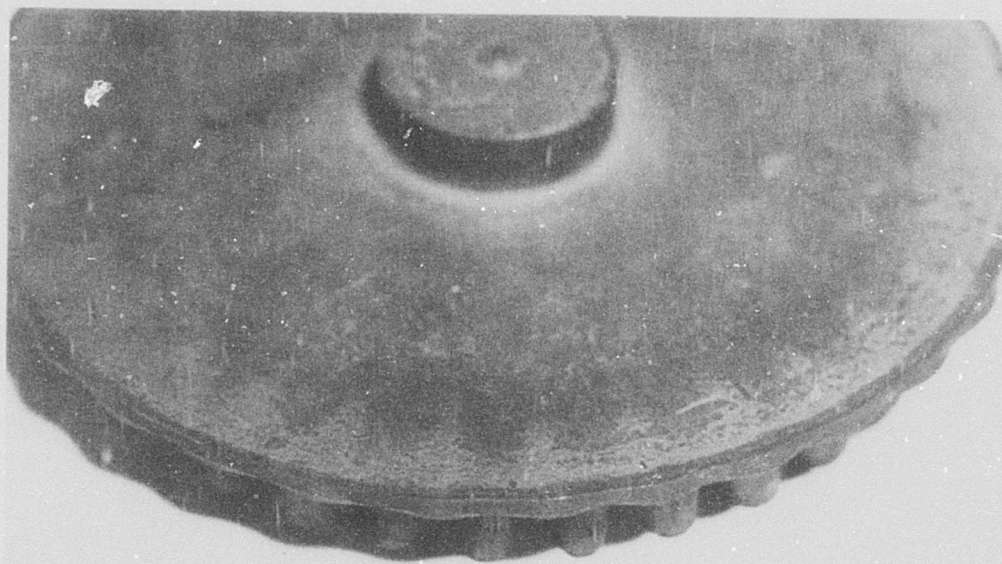
Process "C"

The process "C" forging was produced by two immediately successive forging hits after an initial heat soak at 1850°F. The energy transferred to the billet during the initial forging operation maintained the billet temperature at approximately 1850°F for the second forging operation. This gear blank is shown in Figure 6.

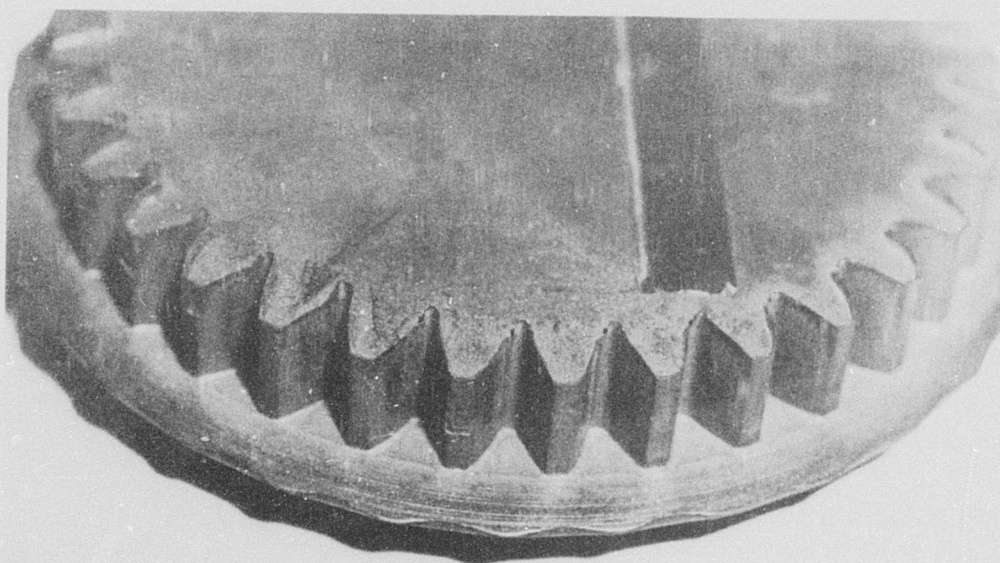
All the gear blanks received by Sikorsky conformed to the .010/.014-inch material allowance requirement. It can be seen by the flash obtained in the final as-forged condition that each forging source used a different die design. In spite of the differences in dies and in basic metal-working procedures, each process produced acceptable gear blanks. Table I outlines the pertinent forging data for each process.



Figure 3. Conventional Pancake Forging.

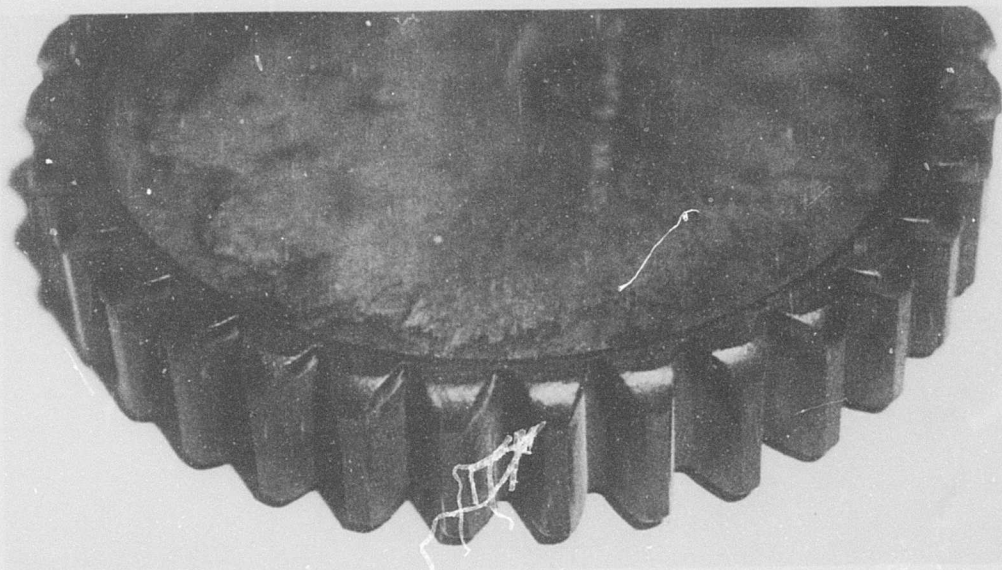


Top View

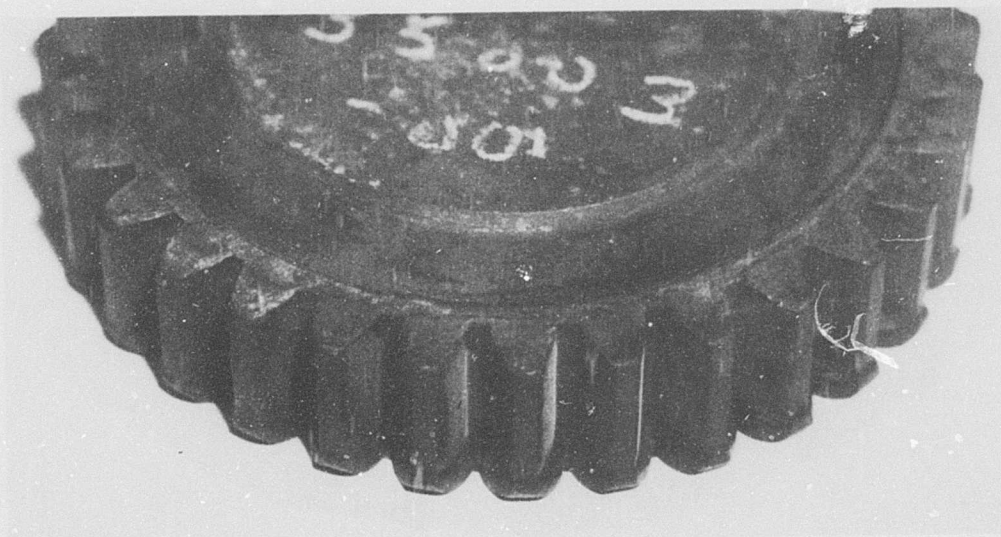


Bottom View

Figure 4. Process "A" As-Forged Gear Blank.

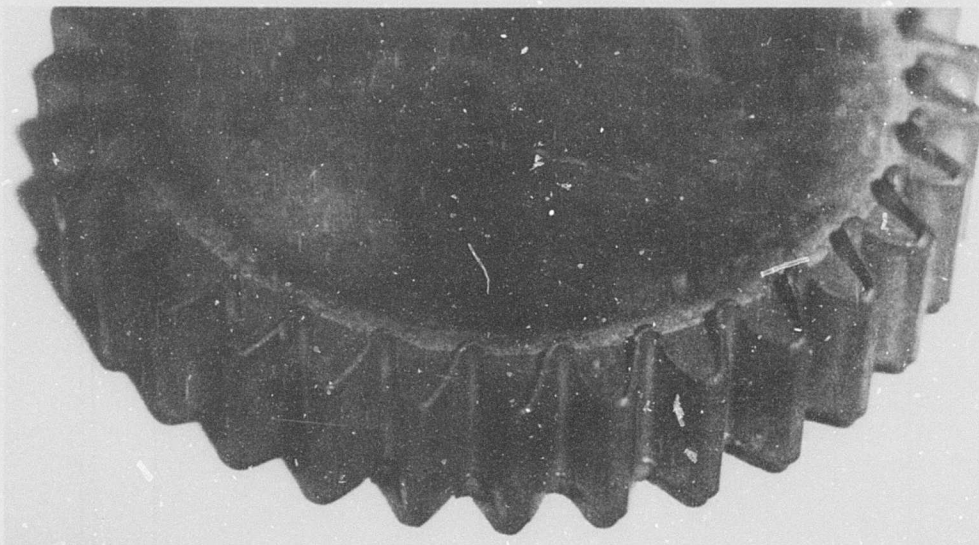


Top View



Bottom View

Figure 5. Process "B" As-Forged Gear Blank.



Top View



Bottom View

Figure 6. Process "C" As-Forged Gear Blank.

TABLE I. BASIC FORGING DATA				
Process				
	"A"	"B"	"C"	PAN
Initial Billet Size, in.	2½ dia. x 1.81	2½ dia. x 1.42	2.125 dia. x 2.50	2.50 dia. x 2.50
Number Forging Hits	3*	1	2	-
Forging Temperature First Hit, °F	1,850	2,100	1,850	2,200
Intermediate Billet Size, in.	-	None	3.6 dia. x .95	None
Intermediate Processing	Grit blast and turn to 3.780 in. dia.	-	None	-
Final Forging Temperature, °F	1,775	-	1,850	-
Forging Press	Pneumatic-mechanical	Pneumatic-mechanical	Mechanical	Steam hammer
* Intermediate Billet Produced by Two Blows				

GEAR MANUFACTURE

DISCUSSION

The procedure that was followed throughout the dynamic test program and particularly during the processing of the test specimens was such as to minimize or eliminate many of the variables that could cause experimental error. Consequently, within each lot, the test gears were machined and ground in a random order to minimize the effect of such variables as machine drift, tool wear, and machine operator. Gears in both lots were manufactured at the same manufacturing facility (Fenn Manufacturing, Newington, Conn.) to insure common heat-treating and machining practices. A gear processing program, Table II, was supplied to the manufacturer to control the manufacturing sequence and randomization of the gear blanks.

TEMPERING

As each gear blank lot was received, it was identified and serialized. All forgings were received with a hardness ranging from R_c 21 to R_c 28. To facilitate machining and avoid the green-grinding difficulties experienced in the previous single-tooth program, where constant redressing of the grinding wheel was required, all gear blanks were hardened and tempered to R_c 33-35 prior to machining.

MACHINING

The first machining phase consisted of rough-machining one side, the inside and outside diameters, and the undercut. The serial number was immediately stamped in the machined recess for permanent identification. The second side was then rough-machined. At this point the pancake forgings were hobbed to the tooth dimensions of the advanced forging blanks. All gears were then drilled and reamed (with allowance made for final grinding) to form the four attachment holes. The gear blanks were then intermixed in a systematically arranged sequence for green-grinding.

GREEN-GRINDING

The test gears were green-ground (except where noted) to remove approximately 0.003 inch from the tooth profile before heat-treating.

Inspection following the green-grinding showed that random small areas on the tooth flanks and top land of the advanced gear forgings failed to clean up completely, indicating that some random tooth-spacing errors existed in the as-forged gear blanks.

Detail inspection of the gear blanks during fabrication also revealed that the as-forged tooth profiles were sufficiently accurate to attempt to grind to the final gear tooth profile without any intermediate grinds. Accordingly, two forgings of each advanced process were carburized immediately after the preliminary blanking operations and hard-ground to

establish the final tooth dimensions, thereby eliminating the green-grind operation. Elimination of the green-grinding is important for two reasons: first, the preliminary drawing and tempering operation could also be eliminated from the processing since its purpose is to prevent the otherwise soft metal from smearing and clogging the pores of the grinding wheel during green-grinding; and second, a single grinding operation after heat-treating would mean that less of the beneficial grain flow on the tooth surface produced by the forging operation would be removed during processing, provided, of course, that a smaller material allowance for grinding is designed into the dies. The green-grinding sequence was eliminated from the processing of the gears of Heat 2.

CARBURIZATION

Upon completion of the green-grinding operation, all test gears which were green-ground and the gears from each process for which the green-grinding was eliminated were carburized and heat-treated in a single lot to produce a 0.035-to-0.045-inch depth of case with an equivalent core hardness of R_c 34-40.

FINAL GRINDING

The final grinding operation was accomplished on a Detroit grinder using a 6-inch-diameter wheel. The gear lots were again systematically randomized in a new grinding sequence and ground on a common machine.

Profile charts showing deviations from a true involute form were taken for each test gear. Sample charts for the as-forged gear blank, gears processed with green-grinding, and gears hardened and ground immediately after blanking are included as Figures 7 through 9.

In the previous single-tooth test program of Reference 2, the tips of some of the teeth of the advanced forging processes failed to clean up completely. In this program, all teeth were found to have cleaned up after the final grind. This is probably due to the fact that the dynamic test gear required more tip relief than the single-tooth test gear. The finished gear is shown in Figure 10.

TABLE II. ADVANCED GEAR FORGING PROCESSING PROGRAM

Step	Process	Purpose	Lot Size	Control
1	Serialize forgings	Identification	Individually	Serialize upon receipt of forgings
2	Normalize (air cool to 150°F or less). Harden and temper to R _c 33-35 (350°F min. tempering temperature).	Machinability and structural refinement	All together	
3	Blank	Machine one side, ID and OD, and undercut. Reserialize*. Machine second side.	Individually	Random order
4	Shape pancake forgings to within 0.007 to 0.010 inch of finished size.	Form gear teeth	Individually	Random order
5	Drill and ream (allow for grinding)	Form holes	Individually	Random order
6	Green-grind teeth	Preheat grinding	Individually	Grind in randomized sequence
7	Magnaflux	Grinding inspection	Individually	100% inspection
8	Burr and buff	Remove sharp edges	Individually	Random order
9	Carburize, harden and draw	Normal heat treat procedures	All together	

TABLE II - Continued			
Step	Process	Purpose	Lot Size Control
10	Finish grind (other than teeth)	Establish final dimensions	Individually Random order
11	Jig-grind holes	Finish holes to size	Individually Random order
12	Grind gear teeth	Establish final tooth dimensions	Individually Grind in randomized sequence
13	Nital etch	Inspection for grinding burns	Individually Random order
14	Burr and buff	Remove sharp edges	Individually Random order
15	Magnaflux	Grinding inspection	Individually 100% inspection
16	Parco-Lubrizo finish	Protective coating	All together
17	Magnaflux and inspect visually	Final check	Individually 100% inspection
<p>* Serialization must be accomplished immediately after undercutting.</p> <p>NOTE: All gears will be dimensionally inspected and over-pin dimensions will be recorded and submitted with the finished gears. Involute profile and lead charts for each test gear will also be provided.</p>			

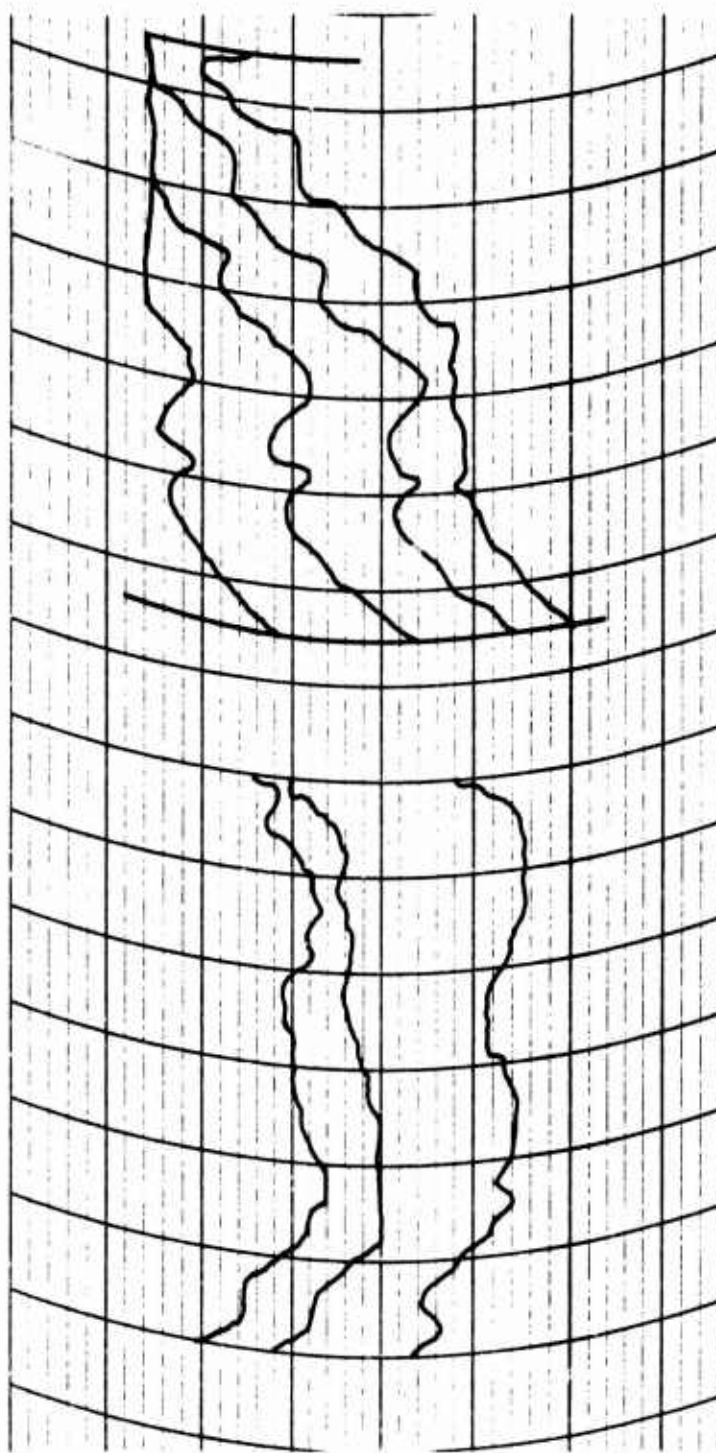


Figure 7. Involute Profile Chart, Typical As-Forged Gear Blank.

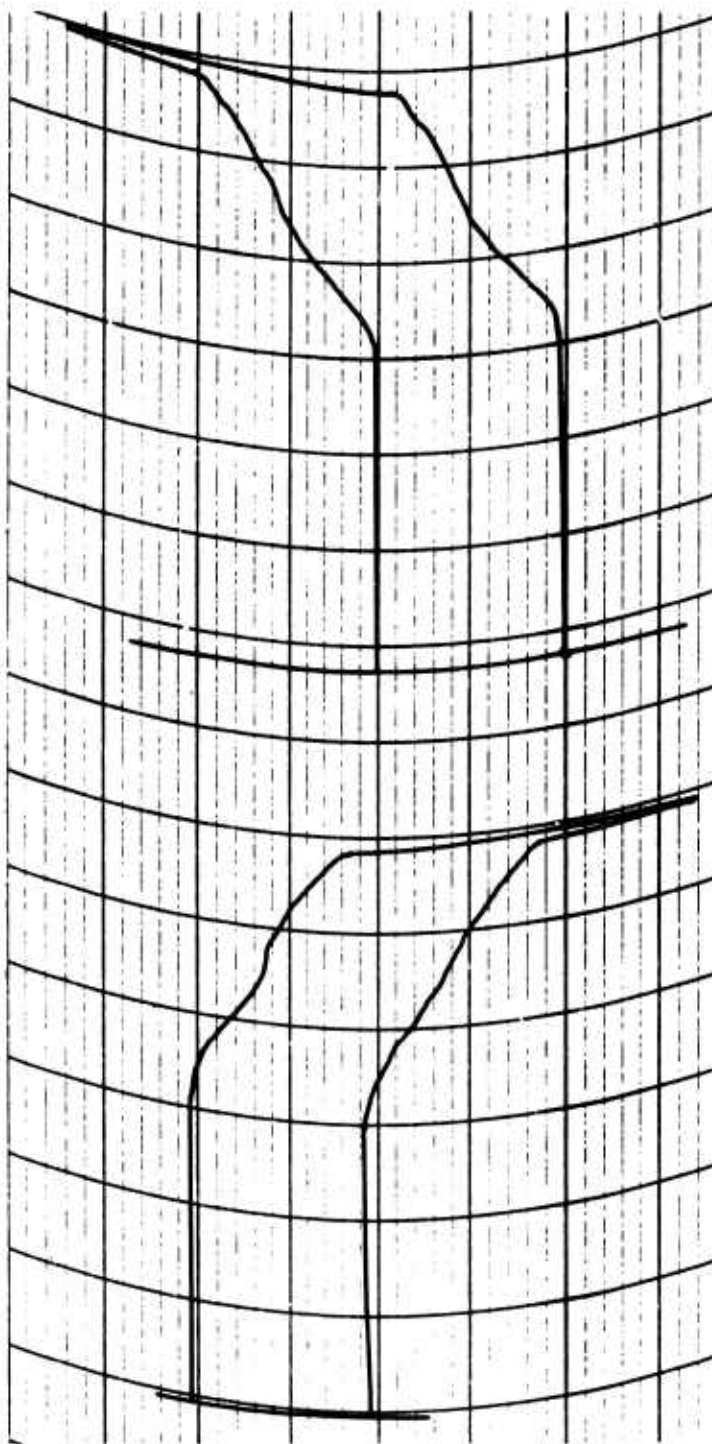


Figure 8. Involute Profile Chart,
Gears Processed With
Green-Grinding, S/N F17.

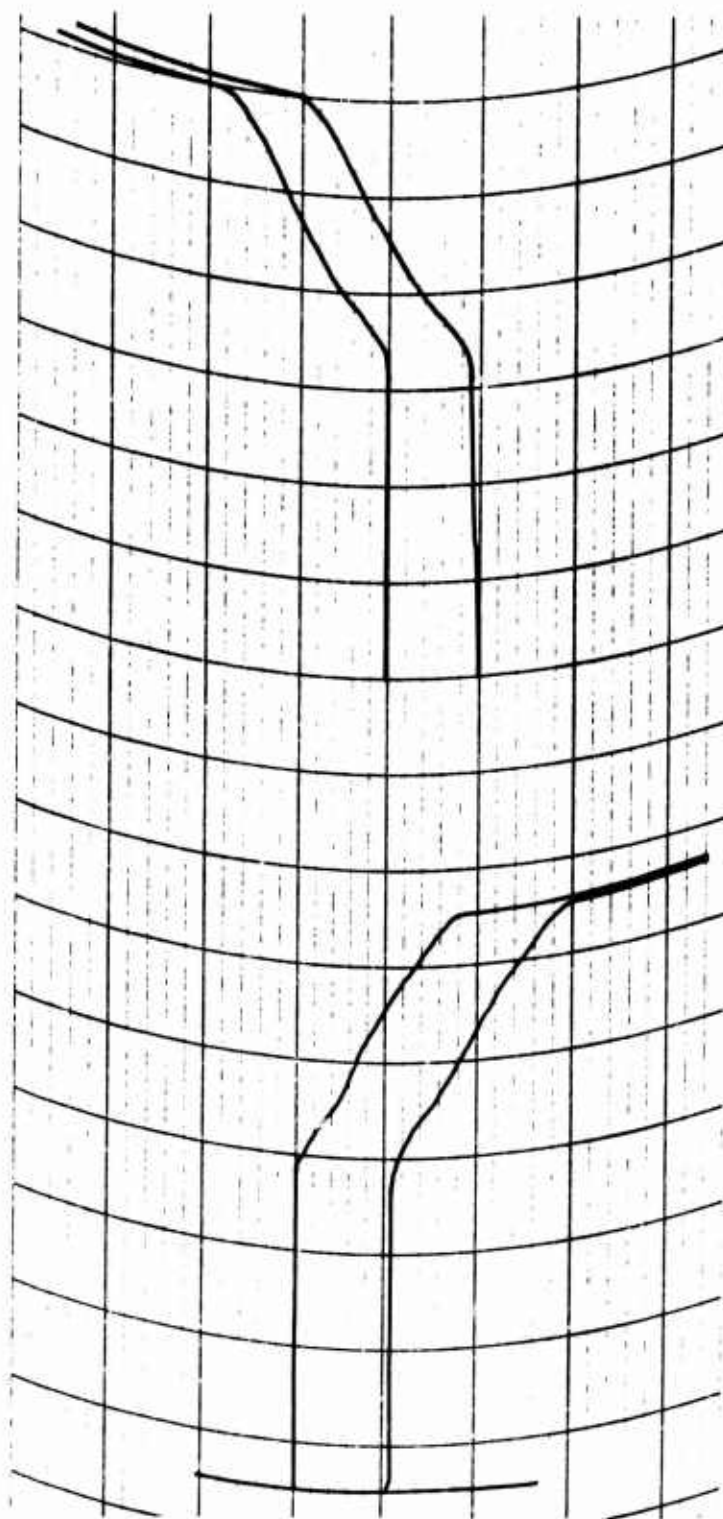


Figure 9. Involute Profile Chart,
Gears Processed Without
Green-Grinding, S/N F27.

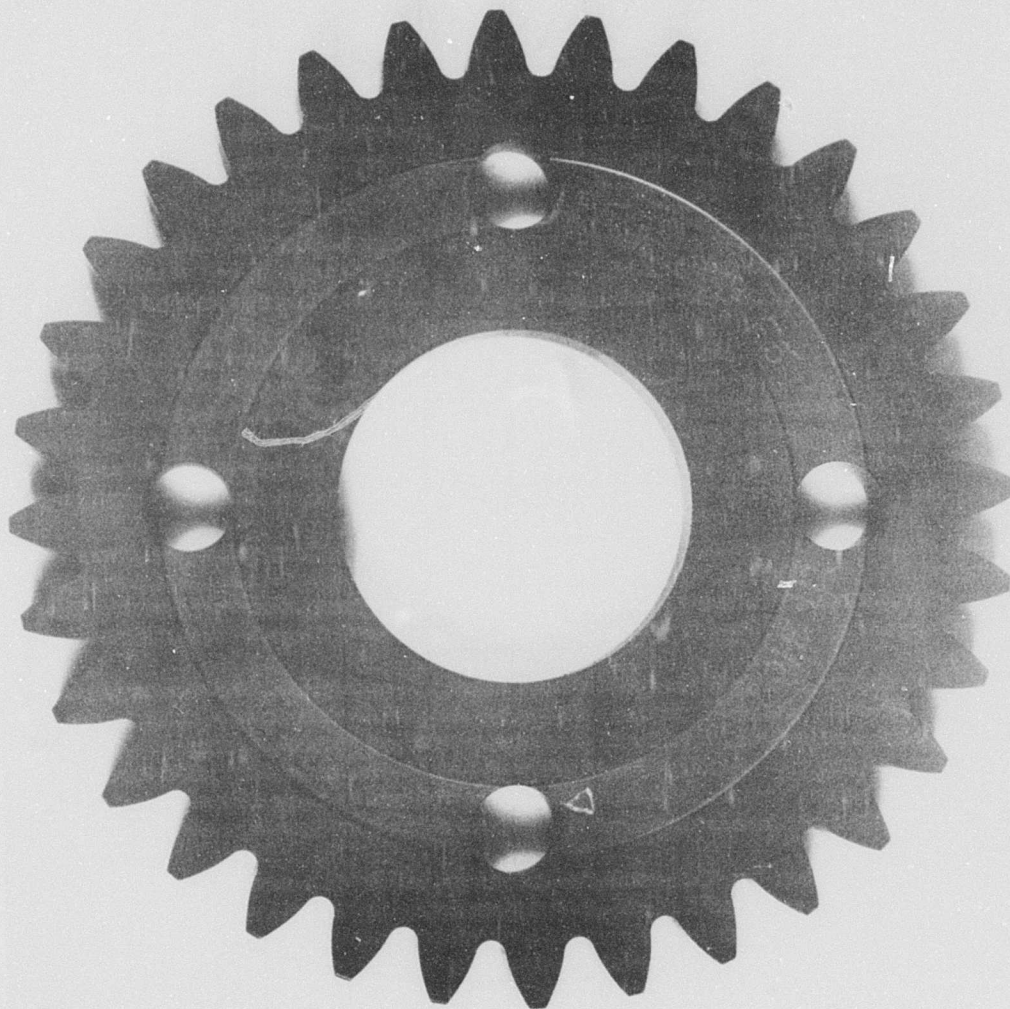


Figure 10. Finished Test Gear.

TEST PROCEDURE

TEST FACILITY

A Sikorsky-designed test facility, incorporating a four-square closed-loop regenerative test rig, Figure 11, was used to evaluate the test gears. In this test facility, two gearboxes, each containing two test gears, are connected by shafts mounted on flexible couplings to reduce interactions between the two gearboxes. This arrangement permits four gears to be tested simultaneously.

To facilitate replacement and inspection, the gears are piloted on the outboard end of the gearbox shafts. Torque is transmitted to the gears by four close-tolerance bolts which also retain the gears on the shafts.

The original configuration of this test facility incorporated an idler gear between the two test gears in each gearbox. The present improved configuration has two pairs of test gears at each end mounted in a staggered arrangement as shown in the schematic of Figure 11. This eliminates reversed bending loads on the idler gear and prevents progressive secondary fractures when a single test gear fails.

A 40-horsepower, 1750-rpm electric motor supplies the necessary power to overcome the friction of the system. A vee-belt drive with a 1.75 to 1.0 pulley ratio transmits the power to the test gearbox. A spur gear set with a 3.3 to 1.0 ratio delivers the power to the closed loop at 9200 rpm.

Torque is applied to the system by the relative angular displacement of verrier plates on one of the connecting shafts. System "wind-up" provides adequate sensitivity to obtain the desired torque levels. Strain gages bonded in a torque-sensing orientation to the connecting drive shafts and wired into a Wheatstone bridge configuration are used to measure system torque while the load is being applied.

Temperatures are measured using iron-constantan thermocouples on oil-in and oil-out lines and on bearing housings in close proximity to each shaft bearing (twelve locations in all) to constantly monitor gearbox bearing temperatures. These bearing temperatures are continually recorded on a Speedomax recorder. A Cox flow turbine in each oil input line measures oil flow rates. Pump output and last-jet oil pressures are measured to monitor pump operation and gearbox pressure drops.

Each gearbox has an independent lubrication system operating at flow rates of up to 5 gpm. Each oil reservoir has a capacity of 15 gallons of oil. The maximum heat rejection capacity for each cooling system is 50,000 Btu per hour. A 40-micron filter on each supply line maintains oil cleanliness and prevents oil jet blockage.

A failure detection system which would automatically shut down the test facility when a failure occurred is installed as part of the test facility

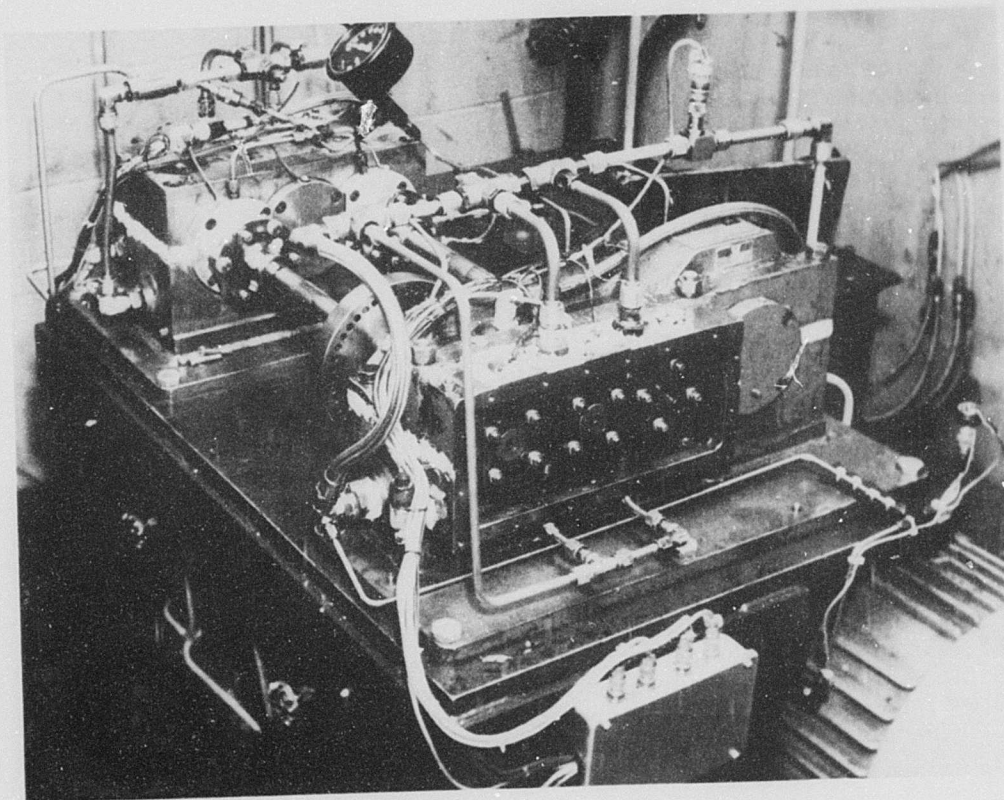
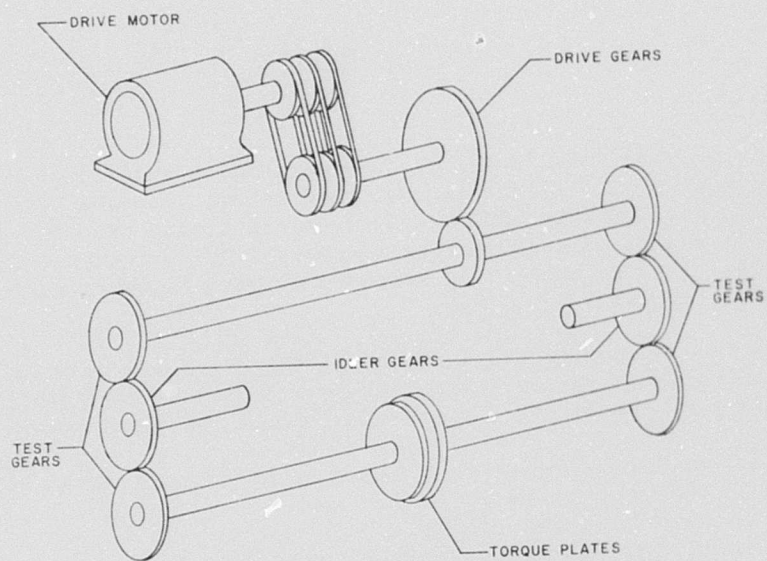


Figure 11. Sikorsky Dynamic Test Facility.

circuitry. A low oil pressure switch protects the facility from failures due to malfunctioning oil pumps, ruptured oil lines, or low oil level in either sump. Excessive oil temperature also activates the shutdown system. Magnetic type chip detectors are incorporated to stop the test if metallic particles enter the lubrication system. The possibility of a vibratory shutoff system was investigated, but it was found that the failure would have to progress to the point where secondary damage to adjacent teeth on both test gear and idler occurred before the shutdown mechanism was activated. The best indication of impending tooth failure was found to be an audible pitch change in combination with a noticeable gearbox temperature rise.

In an effort to further improve the failure detection system a unique feature, in the form of a missing tooth detector, was added prior to the initiation of the test series of Heat 2. This device compares an input signal from a magnetic tooth contactor (on a cycle per cycle basis) to an internal signal generated by an oscillator which is phase-locked to the contactor signal. If a tooth is missing, the comparison on that cycle will trigger a flip-flop which will trip the motor relay to shut off the machine. The time from detection to relay shutoff is approximately equal to the relay closing time. There is one circuit for each of the eight gear positions. When a bending failure occurs, which results in the loss of a gear tooth, lights on the instrument panel will indicate not only which gearbox is affected but in which of the four possible gear positions the failed gear can be found. If the test machine shuts down because of low oil pressure, low oil level, chip detection, or a recorder malfunction (high temperature), this fact will also be indicated by an appropriate light on the instrument panel.

FATIGUE TESTS

The dynamic fatigue tests on Heat 1 were conducted in a randomized order on nine gears of each forging type. Three gears of each process were tested at each of three load levels for a total of thirty-six test points. Runout was established at 30 million cycles.

The initial test run was made at a tangential tooth load of 1000 pounds. When all four test gears reached 30 million cycles at this load without failure, it was decided to increase the minimum test load to 1250 pounds and adjust the higher loads accordingly. The final gear tooth loads and corresponding gear tooth stresses (calculated using the methods of Reference 3) used throughout the remainder of the test program are shown in Table III.

TABLE III. TOOTH LOADS AND STRESSES, HEAT 1		
Tooth Load (lb)	AGMA Bending Stress (psi)	Compressive Stress (psi)
1,250	64,200	222,500
1,750	88,900	264,200
2,250	111,300	299,600

The tests of Heat 2 were similarly conducted on 12 gears of each process (the conventional process and advanced forging process "A") for a total of 24 additional test points.

The gear tooth loads used and the corresponding gear tooth stresses are shown in Table IV.

TABLE IV. TOOTH LOADS AND STRESSES, HEAT 2		
Tooth Load (lb)	AGMA Bending Stress (psi)	Compressive Stress (psi)
1,500	76,200	244,600
1,750	88,900	264,200
2,000	101,600	282,500
2,250	114,300	299,600

Since the design of the test facility permits four gears to be tested at one time, the position of each gear in the tester was randomized in each test series to preclude the possibility of error due to gear location. When a gear failure occurred in a test run, all test gears were removed from the tester and tagged with a notation of the test run, test load, and number of cycles accumulated in that run. The gears which did not fail were then stored for a later run at the same load level. This procedure was followed for each test run until failure or runout of all test gears occurred.

The test gears were identified by serial number as shown in Tables V and VI. The test gears which were specially processed during manufacture to eliminate the green-grinding operation are serial numbers F01, F02, F12, F13, F23, and F27 of Heat 1, and all of the gears of Heat 2.

TABLE V. GEAR SERIALIZATION, HEAT 1		
Forging Technique	Part Number	S/N
Process "A"	6105-35033-105	01-10
Process "B"	6105-35055-106	11-20
Process "C"	6105-35033-107	21-30
PAN	6105-35033-108	31-40

TABLE VI. GEAR SERIALIZATION, HEAT 2		
Forging Technique	Part Number	S/N
Process "A"	61050-35080-101	01-40
PAN	61050-35080-102	41-80

TEST RESULTS

TEST DATA

Heat 1

The first set of gears in the dynamic fatigue test were run at a tooth load of 1000 pounds, and all ran out to 30 million cycles without failure. Inspection of the test gears and idlers at the completion of this run revealed signs of minor interference at the tip of the teeth and also in the region of the TIF diameter, suggesting that the test gears might require additional tip relief. The same test gears were rerun on the reverse side at 1750 pounds. This test was terminated at one million cycles due to severe scoring on the tips of the teeth, thus substantiating the need for additional tip modification.

All test gears and idlers were returned to the gear manufacturer for re-grinding to increase the tip relief from $-.0003/-0.0007$ to $-.0014/.0019$ inch. The remainder of the test runs were made with the modified gears at the test loads of Table III. Runout was established at 30 million cycles. The fatigue test data are summarized in Table VII.

As shown in Table VII, the two sets of gears tested at the 1250-pound load level ran out to 30 million cycles without failure. These gears were subsequently retested at a test load of 1500 pounds in an attempt to better define the high-cycle region of the S-N curve. These points are identified in Figures 12 through 15.

Heat 2

Based on the observation that some of the test gears of Heat 1, particularly those tested at the high load levels, still showed signs of tip scoring at the higher tip relief, it was decided to again increase the tip relief on the test gears by .0003 inch. Initial testing at the 1500-pound load level produced 3 failures, all in less than 500,000 cycles. A failure also occurred at 1750 pounds in 560,000 cycles. Since these times are considered low for this gear, based upon previous testing, a metallurgical check of the test gear and a load calibration check of the loading mechanism were made but no abnormalities were noted. It was concluded, then, that the early failures were associated with the high tip relief which, while beneficial at the higher loads where tooth deflections need compensation, could possibly precipitate early failures at the low loads due to high concentration of load at the pitch line. It was decided at this point to return approximately half of the test gears to the manufacturer to reduce the tip relief back to $-.0014/-0.0019$ inch. These reworked gears were used only in the two low load levels. Test results with the reduced tip relief were satisfactory. The test data are summarized in Table VIII.

TABLE VII. TEST RESULTS, HEAT 1

Test Run	Test Load (lb)	Identification	S/N	Total Accumulated Cycles x 10 ⁶	Comments
1	1000	"A"	F02	30.0000	Tip Scoring
	1000	"B"	F19	30.0000	on all
	1000	"C"	F22	30.0000	gears
	1000	PAN	F32	30.0000	
2	1750	"A"	F02	1.0000	Tip Scoring
	1750	"B"	F19	1.0000	on all
	1750	"C"	F22	1.0000	gears
	1750	PAN	F32	1.0000	
3	2250	"A"	F05	.1652	Tooth Fracture
4	2250	"B"	F14	.3308	Tooth Fracture
5	1750	"C"	F26	15.7000	Tooth Fracture
6	1250	"A"	F09	30.0000	Runouts - no
	1250	"B"	F13	30.0000	failures
	1250	"C"	F29	30.0000	
	1250	PAN	F34	30.0000	
7	2250	PAN	F31	.1840	Tooth Fracture
8	1250	"A"	F01	30.0000	Runouts - no
	1250	"B"	F18	30.0000	failures
	1250	"C"	F24	30.0000	
	1250	PAN	F33	30.0000	
9	2250	"A"	F07	.0920	Idler gear
	2250	"B"	F17	.0920	failed
	2250	"C"	F25	.0920	
	2250	PAN	F36	.0920	
10	1750	"C"	F28	8.1512	Tooth Fracture
11	1750	PAN	F37	8.2610	Tooth Fracture
	1750	"C"	F30	.1140	Secondary Failure
12	1750	"A"	F04	18.8508	Tooth Fracture
13	1750	"B"	F12	23.9016	Tooth Fracture
	1750	PAN	F38	5.0508	Secondary Failure
14	1750	"A"	F08	16.3024	Tooth Fracture
15	1750	PAN	F39	16.528	Tooth Fracture
16	1750	"A"	F03	1.5916	Tooth Fracture
17	1750	"B"	F11	15.9938	Tooth Fracture
18	1750	"B"	F15	17.6456	Tooth Fracture
19	2250	PAN	F36	.3596	Tooth Fracture
	2250	"A"	F07	.3596	Tooth Fracture
20	2250	"B"	F17	.5528	Tooth Fracture
21	2250	"A"	F10	.2392	Tooth Fracture
22	2250	"B"	F16	.3956	Tooth Fracture
23	2250	"C"	F25	.6080	Tooth Fracture
24	2250	PAN	F35	.5884	Tooth Fracture

TABLE VII - Continued					
Test Run	Test Load (lb)	Identification	S/N	Total Accumulated Cycles $\times 10^6$	Comments
25	2250	"C"	F23	.5240	Tooth Fracture
26	2250	"C"	F21	.6072	Tooth Fracture
27	1500	"A"	F09	27.5080	Tooth Fracture
28	1500	PAN	F34	27.7100	Tooth Fracture
	1500	"C"	F29	27.7100	Secondary Failure
29	1500	"B"	F13	28.7229	Tooth Fracture
30	1500	"A"	F01	1.1960	Tooth Fracture
31	1500	"C"	F24	1.4904	Tooth Fracture
32	1500	"B"	F18	1.0120	Tooth Fracture

TABLE VIII. TEST RESULTS, HEAT 2

Test Run	Test Load (lb)	Identification	S/N	Total Accumulated Cycles x 10 ⁶	Comments
1	1500	"A"	F20	.3588	Premature Failure
2	1500	PAN	F61	.4324	Premature Failure
3	1500	"A"	F04	.4950	Premature Failure
4	1750	"A"	F31	.5612	Premature Failure
5	2000	"A"	F05	.3596	Tooth Fracture
6	2000	"A"	F03	.5060	Tooth Fracture
7	2000	"A"	F17	.1932	Tooth Fracture
8	2000	PAN	F41	.7084	Tooth Fracture
9	2000	PAN	F58	.2944	Tooth Fracture
10	2000	"A"	F19	.2852	Tooth Fracture
11	2000	PAN	F51	.8464	Tooth Fracture
12	2000	"A"	F02	.2944	Tooth Fracture
13	2000	PAN	F48	.2576	Tooth Fracture
14	2000	"A"	F30	.5980	Tooth Fracture
15	2000	"A"	F13	.4876	Tooth Fracture
16	2250	"A"	F01	.1104	Tooth Fracture
17	2250	PAN	F49	.1472	Tooth Fracture
18	2250	"A"	F28	.1720	Tooth Fracture
19	2250	"A"	F18	.1840	Tooth Fracture
20	2250	PAN	F56	.2300	Tooth Fracture
21	2250	PAN	F79	.2300	No Failure
22	2250	PAN	F53	.1748	Tooth Fracture
23	1500	PAN	F69	6.7252	Tooth Fracture
24	1500	"A"	F32	23.0000	Tooth Fracture
25	1500	PAN	F76	11.3436	Tooth Fracture
26	1500	"A"	F27	42.4672	Tooth Fracture
27	1500	"A"	F34	2.0976	Tooth Fracture
28	1500	"A"	F39	44.1968	Tooth Fracture
29	1750	PAN	F66	5.7408	Tooth Fracture
30	1750	"A"	F14	8.3628	Tooth Fracture
31	1750	"A"	F38	2.3552	Tooth Fracture
32	1750	"A"	F23	2.1068	Tooth Fracture
33	1750	PAN	F77	6.4124	Tooth Fracture
34	1750	PAN	F67	1.3524	Tooth Fracture
35	1750	PAN	F70	8.3628	No Failure
36	2000	PAN	F64	1.3524	Tooth Fracture
37	2000	PAN	F73	1.3524	No Failure
38	1500	"A"	F25	46.6164	Tooth Fracture
39	1500	PAN	F72	46.6164	No Failure
40	1500	PAN	F68	46.6164	No Failure

DATA ANALYSIS

The data from the dynamic test programs were analyzed statistically with the aid of a Sikorsky Aircraft computer program. This statistical approach is based on the theory that for a particular process, there exists a stress or load level below which a failure will never occur (endurance limit) no matter how many stress cycles are imposed. In line with this theory the mean stress/life (S-N) curve can be written in the general form

$$S = E + \frac{\beta}{N^\gamma} \quad (1)$$

where S = stress or load level

E = endurance limit

β = material constant

γ = material constant

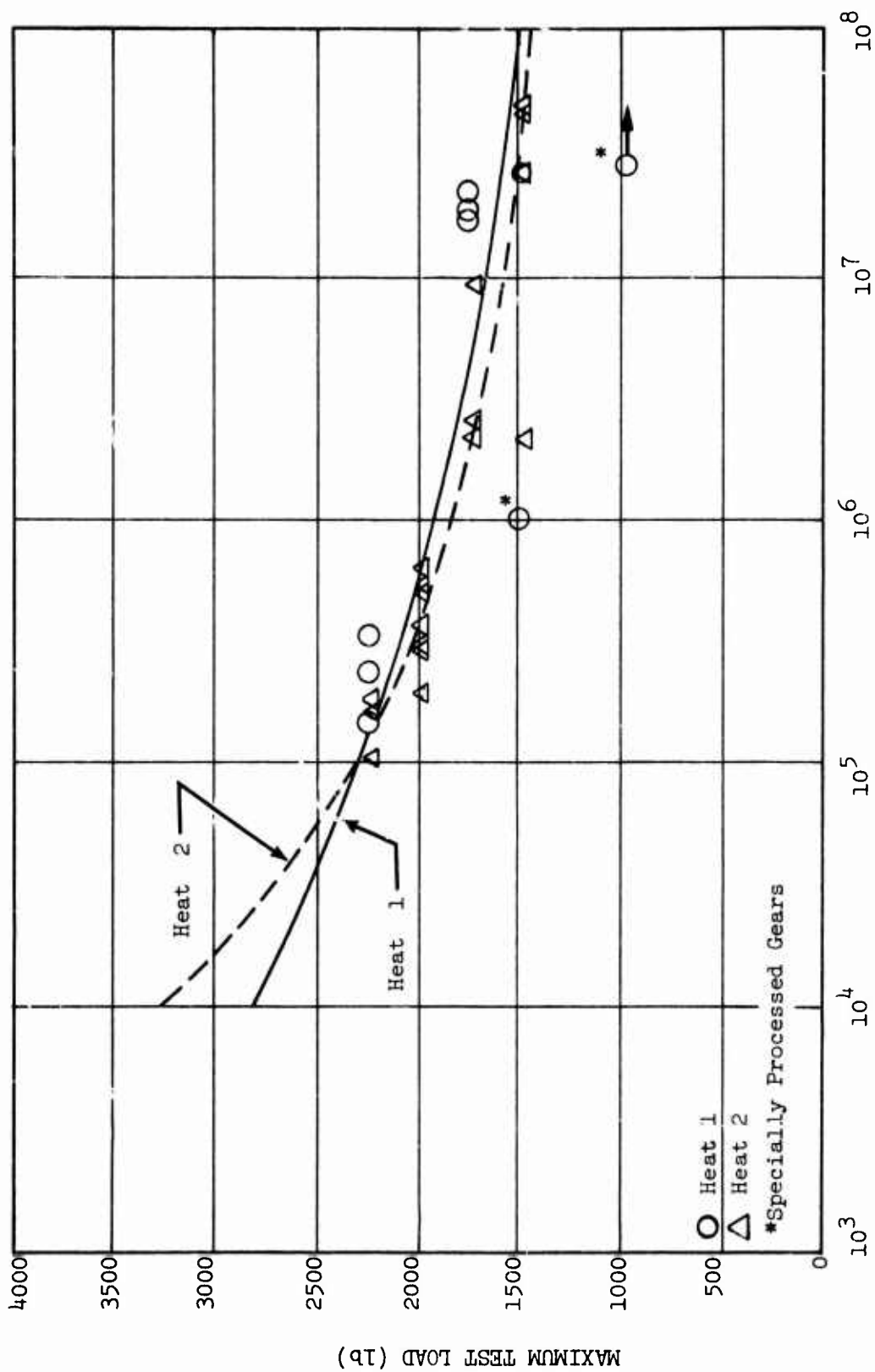
N = cycles to failure $\times 10^{-6}$

A curve equation of this form was used in Reference 4 to plot the results of single-tooth tests on advanced gear materials. A "best fit" S-N curve conforming to Equation (1) was derived from the test data for each process by the computer using the method of least squares. The constants determined by this method satisfy the condition that the sum of the squares of the deviations of stress from the mean curve is a minimum. Using the given set of test data as input, the computer calculates the curve parameters E, β , and γ from a purely objective and unbiased viewpoint, thus eliminating the need for preplotting and curve adjusting.

Due to the limited number of test points in Heat 1 and their grouping, it was not possible to determine independent and realistic values for the three constants of Equation (1). Therefore, to compare the results of the fatigue tests without bias, it was conservatively assumed, from inspection of the test data, that the endurance limit at $N = \infty$ (E in Equation (1)) was equal to 1000 pounds for each process. The remaining constants of Equation (1) were then evaluated using the methods described above. The fatigue limit at 10^8 cycles was then used as the basis for comparing the various processes.

Application of Equation (1) to the test data of Heat 2 resulted in finite and realistic values for all three material constants.

After evaluation of the constants, Equation (1) was used to determine the mean load at various values of N for each process. Figures 12, 13, 14, and 15 are the resulting plots of the data points and respective mean S-N curves. For comparative purposes, a composite of the mean curves is presented in Figure 16.



CYCLES TO CRACK DETECTION

Figure 12. Dynamic Test Results for Gears Forged by Process "A".

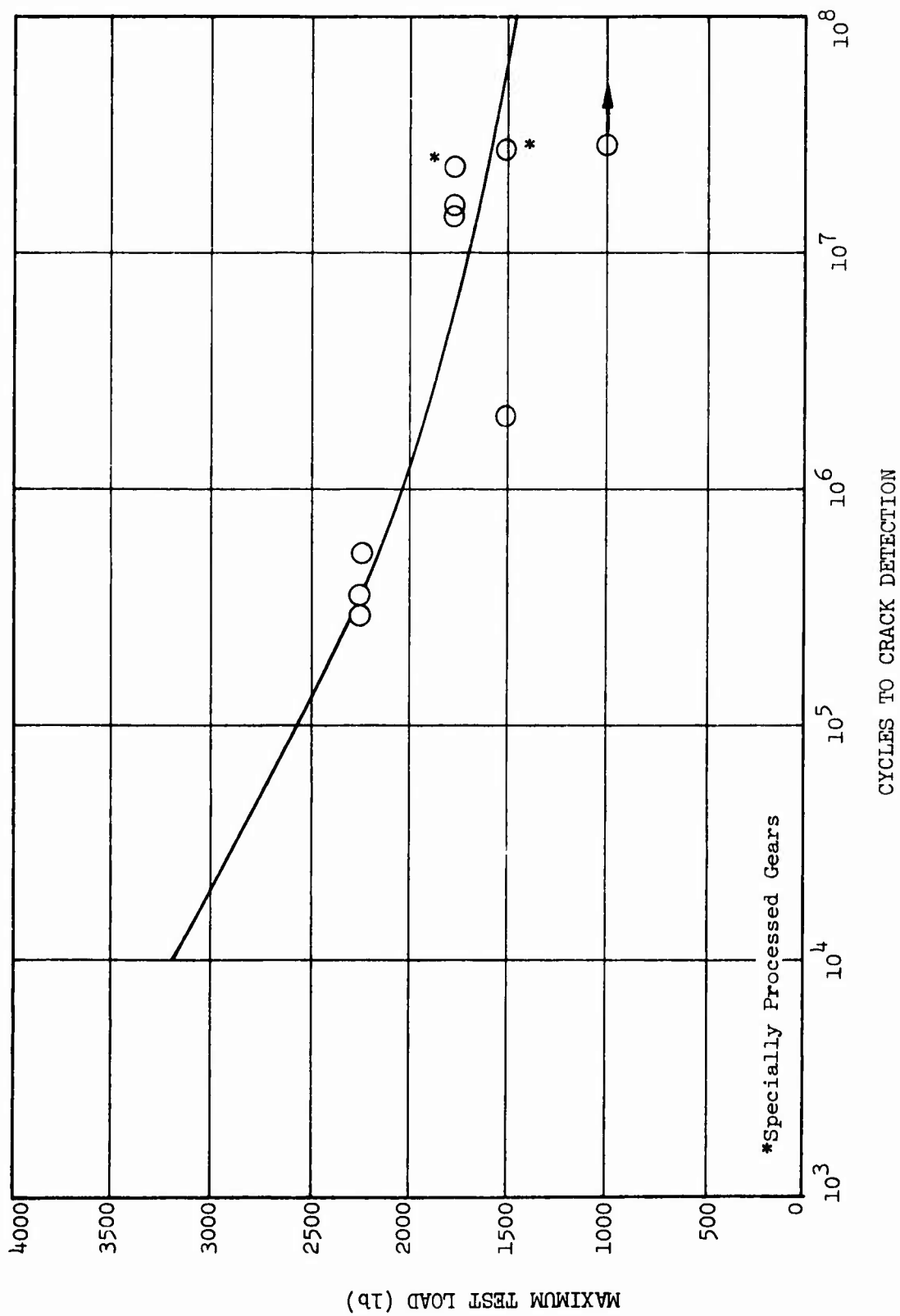


Figure 13. Dynamic Test Results for Gears Forged by Process "B".

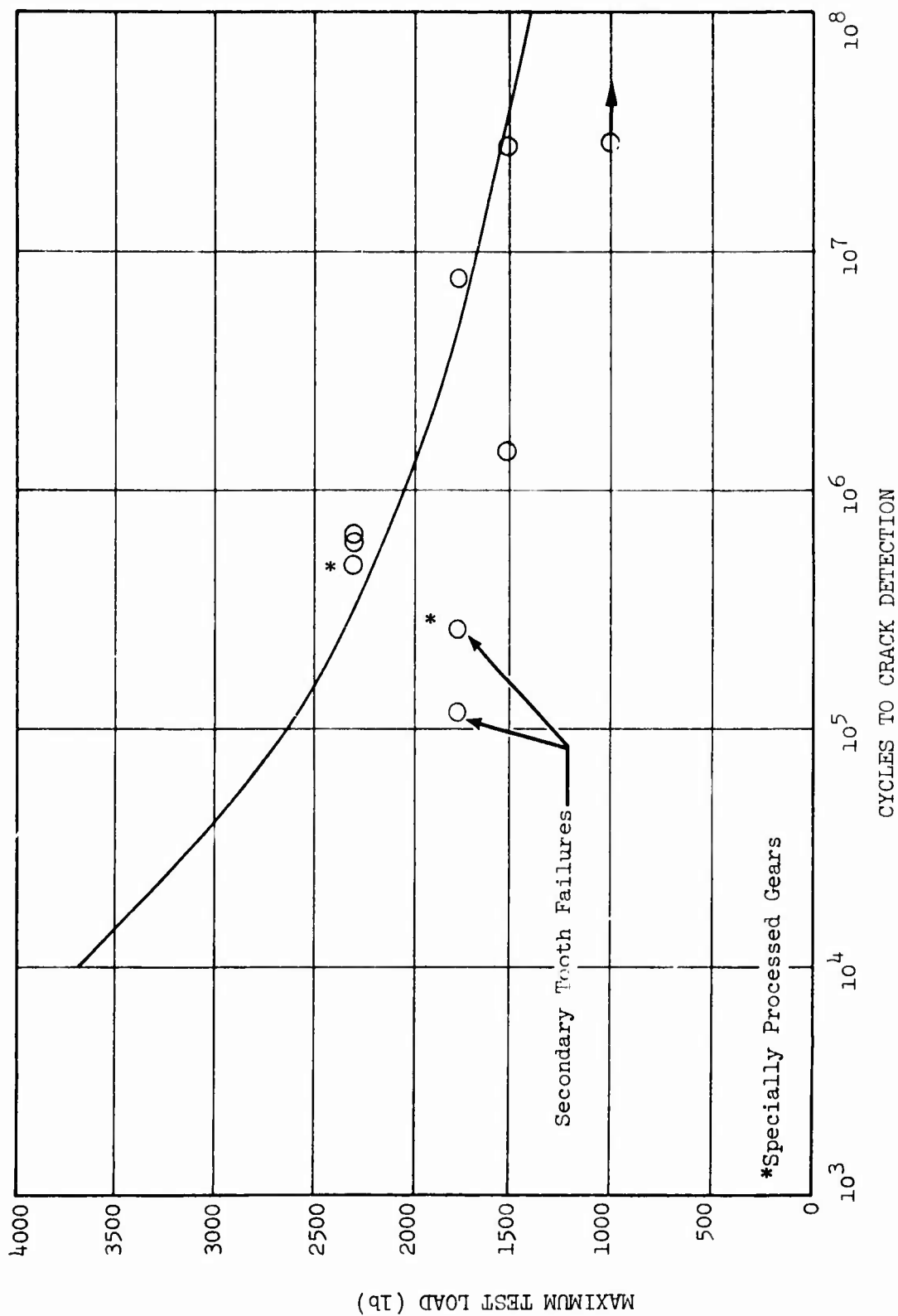


Figure 14. Dynamic Test Results for Gears Forged by Process "C".

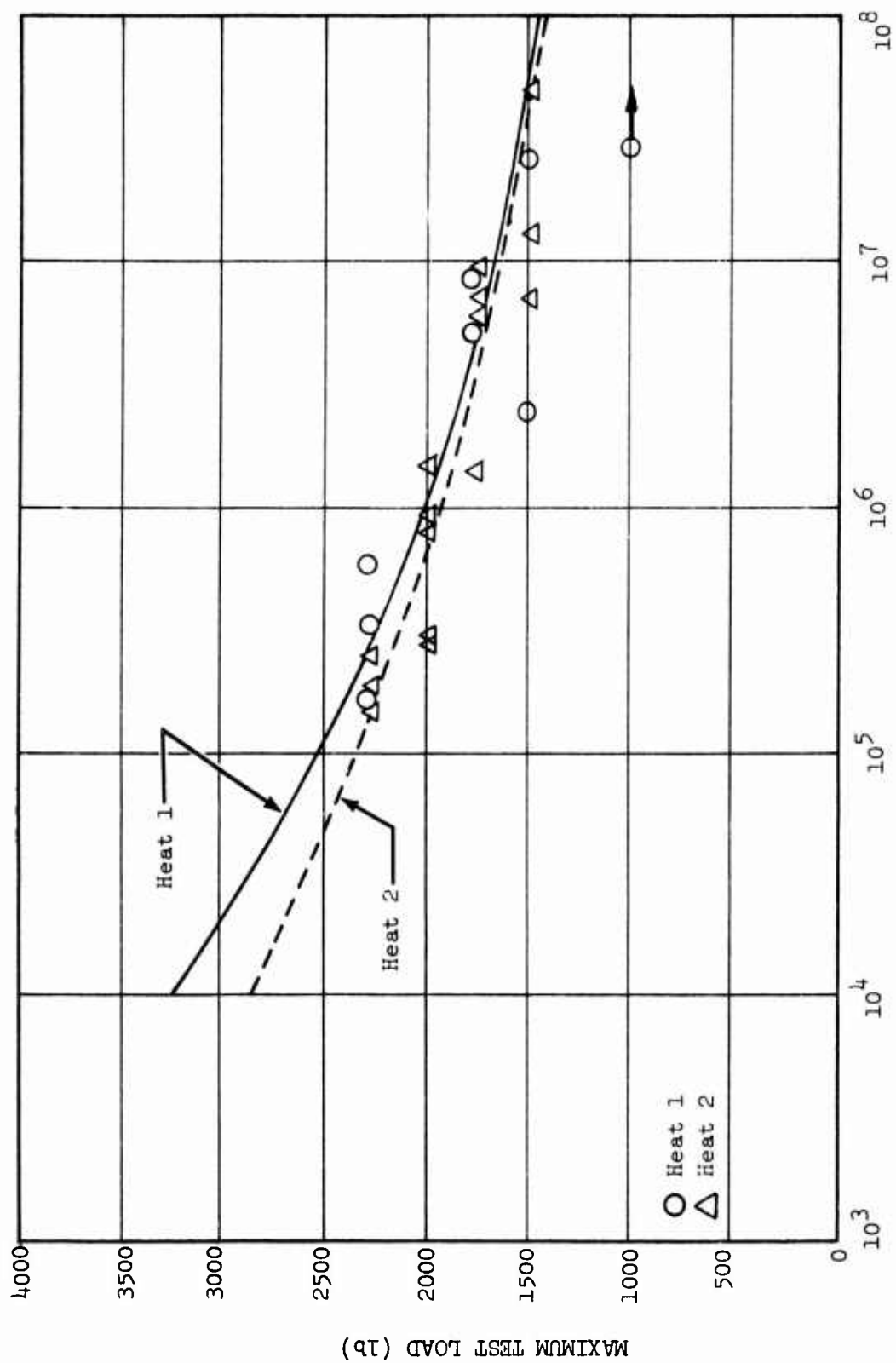


Figure 15. Dynamic Test Results for Gears Forged by Conventional Methods.

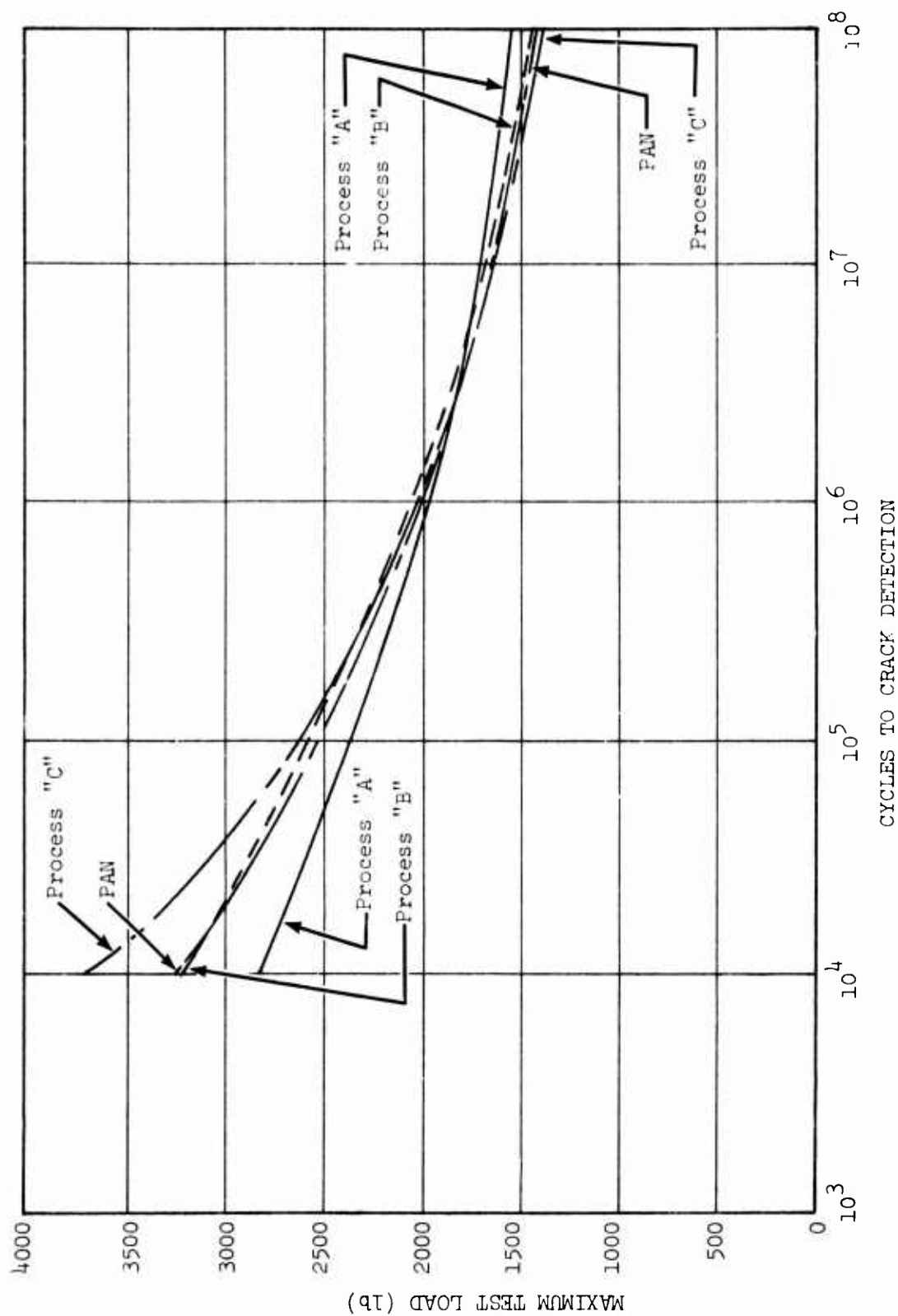


Figure 16. Comparison of Results of Dynamic Fatigue Test.

The distribution of the data about the mean was analyzed statistically by assuming that each individual data point lay on an S-N curve of its own which had the same curve shape as the mean curve for the total group. The mean curve was thus shifted up or down to pass through the specified test point. The value of stress, or load, at 10^8 cycles based on an individual data point was then calculated by

$$\bar{S}_i = \left[\frac{S_i}{1 + \frac{\beta/E}{N_i}} \right] \left[1 + \frac{\beta/E}{100} \right] \quad (2)$$

where \bar{S}_i = equivalent load at 10^8 cycles for each data point

S_i = load at failure for each individual data point

N_i = stress at failure for each individual data point

After the data points were "stacked up" at 10^8 cycles, the standard deviation and coefficient of variation were calculated by

$$s = \sqrt{\frac{\sum (\bar{S}_i)^2 - n(\bar{X})^2}{n - 1}} \quad (3)$$

where s = unbiased standard deviation

\bar{X} = mean fatigue strength at 10^8 cycles for the total group

n = number of test points

and $\frac{s}{\bar{X}}$ = coefficient of variation

The mean fatigue strengths and standard deviations at 10^8 cycles, the coefficients of variation, and the constants of the S-N curve equation for the mean curves are summarized in Tables IX and X.

The probable error between the mean of the test sample at 10^8 cycles and the true mean was calculated using the statistical tables of Reference 6. Based on the probable error of the mean, the spread in the mean fatigue strength at 10^8 cycles for a confidence level of 99 percent was calculated for each process. This data is presented in Tables XI and XII.

TABLE IX. ANALYSIS OF TEST RESULTS, HEAT 1							
Forging Process	No. of Test Points (n)	E (lb)	β	γ	\bar{X}^* (lb)	s ** (lb)	S/ \bar{X} (%)
"A"	8	1000	956	.139	1505	157.2	10.4
"B"	8	1000	1031	.164	1485	144.8	9.8
"C"	7	1000	1032	.201	1411	158.6	11.2
PAN	8	1000	983	.175	1440	128.4	8.9
* Mean Fatigue Strength at 10^8 cycles							
** Unbiased estimate of standard deviation at 10^8 cycles							
Mean Fatigue Strength (%)							

TABLE X. ANALYSIS OF TEST RESULTS, HEAT 2							
Forging Process	No. of Test Points (n)	E (lb)	β	γ	\bar{X}^* (lb)	s ** (lb)	S/ \bar{X} (%)
"A"	18	133+	500	.288	1467	73.6	5.0
PAN	18	592	1343	.110	1402	76.2	5.4
* Mean Fatigue Strength at 10^8 cycles							
** Unbiased estimate of standard deviation at 10^8 cycles							
Mean Fatigue Strength (%)							

TABLE XI. MEAN FATIGUE STRENGTHS AT 99 PERCENT CONFIDENCE LEVEL, HEAT 1						
Forging Process	No. of Test Specimens (N)	s (lb)	Probable Error at 99% Confidence (lb)	Upper * Limit	Mean *	Lower * Limit
"A"	8	157.2	± 194	1699	1505	1311
"B"	8	144.8	± 179	1664	1485	1306
"C"	7	158.6	± 222	1633	1411	1189
PAN	8	128.4	± 159	1599	1440	1281
* Applicable at 10^8 cycles of vibratory stress						

TABLE XII. MEAN FATIGUE STRENGTHS AT 99 PERCENT CONFIDENCE LEVEL, HEAT 2						
Forging Process	No. of Test Specimens (N)	s (lb)	Probable Error at 99% Confidence (lb)	Upper * Limit	Mean *	Lower * Limit
"A"	18	73.6	± 50	1517	1467	1417
PAN	18	76.2	± 52	1454	1402	1350
* Applicable at 10^8 cycles of vibratory stress						

DISCUSSION OF RESULTS

The results of the dynamic fatigue tests shown in Table IX on Heat 1 show that the advanced forging process "A" has a mean fatigue strength at 10^8 cycles that is 5 percent greater than the mean fatigue strength of the conventional pancake forgings, and the mean fatigue strength of forging process "C" is 2 percent less than the conventionally produced gears. Using the lower limit of the 99 percent confidence band at 10^8 cycles, listed in Table VII, these percentages are respectively 2 percent greater and 7 percent less. It is significant to note that the forging process which had the lowest fatigue strength had only 7 failure points compared to 8 for the other processes. One of the gears of this process failed due to secondary damage; therefore, this test point was not used in plotting the S-N curve. The fact that there are fewer test points affects the calculation of the unbiased standard deviation and the probable error of the mean for this process.

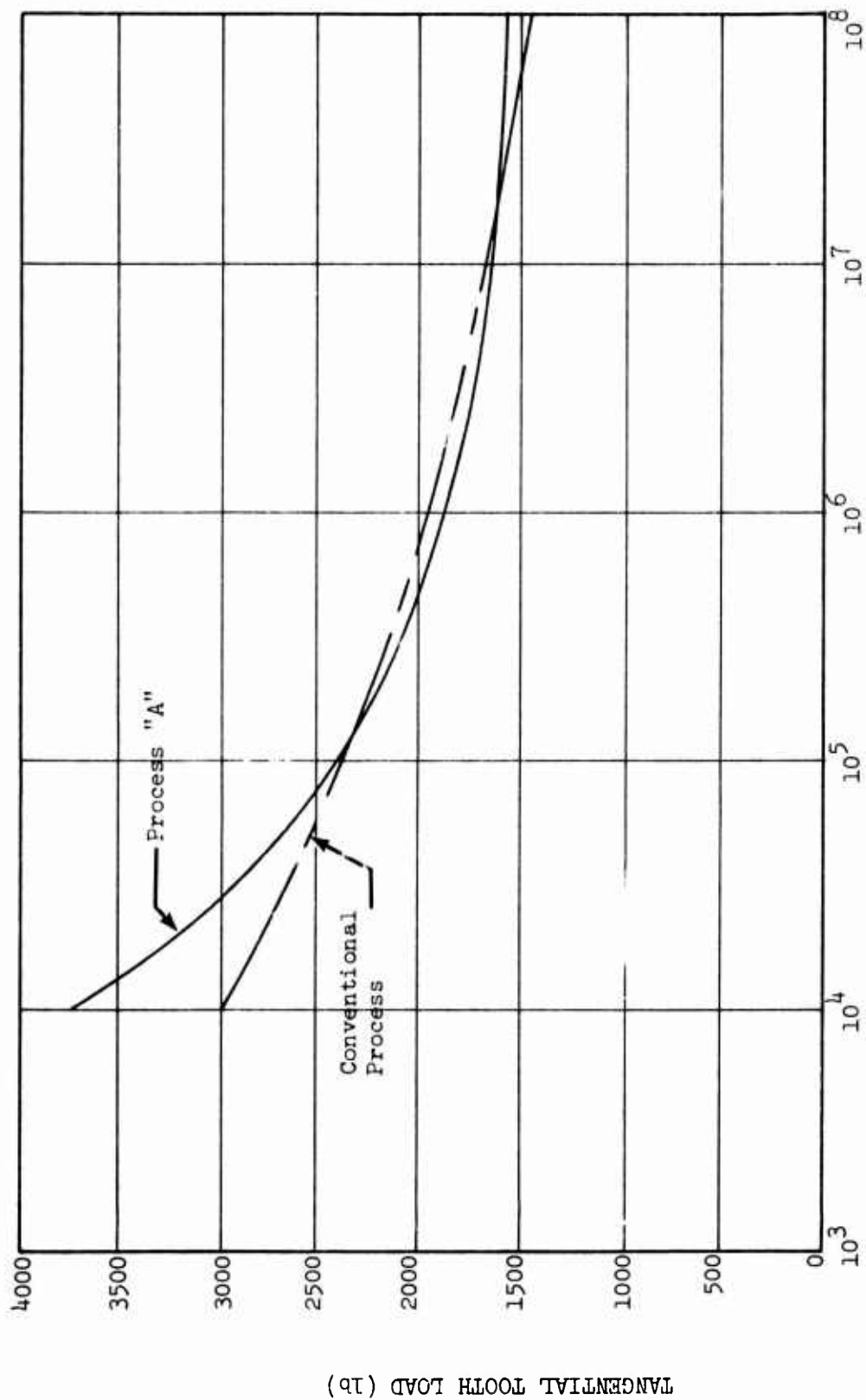
The above results point up the fact that more than three test points at three load levels are required to adequately define the S-N curve relation. Five valid test points (not runouts) at each of four load levels constitute, in the authors' opinion, the minimum number of fatigue test points for a gear fatigue test program of this sort. It was for this reason that additional tests of Heat 2 were initiated.

The test gears which were processed without green-grinding are identified in Figures 12 through 15. Although more test points are required to adequately assess the comparative strengths of these gears, it appears from a review of the data that the special processing had no deleterious effect on their fatigue strength. The test gears of Heat 2 were processed without green-grinding on this basis.

The results of the fatigue tests on Heat 2 (shown in Table X) are almost identical to the results at Heat 1 with regard to the percentage of improvement of the advanced gear forgings (as represented by process "A") over the conventional pancake forging. The actual endurance limit values at 10^8 cycles are, however, about 2 percent lower. The curve shapes, as shown in Figures 12 and 15, for the tests of both heats are very similar.

To judge the effect of additional test points on the ranking of the two forging processes and the shape of the S-N curves, the test results of Heat 1 and Heat 2 were combined and composite S-N curves were drawn by the methods outlined above. The result of this analysis is shown in Figure 17. The endurance limits at 10^8 cycles for the advanced gear forging process "A" and for the conventional process are 1540 and 1432 pounds respectively, which is about an 8 percent improvement. The curve shapes are quite similar to those derived for the individual test lots.

It can reasonably be concluded from the above discussion that the best advanced gear forging process shows an advantage of 5 to 8 percent with respect to gear tooth bending strength at 10^8 cycles, over the conventional forging process.



CYCLES TO FRACTURE

Figure 17. Composite Curves From Heats 1 and 2.

It should be pointed out, however, that due to the diverging character of the S-N curve shapes, this advantage will increase at a higher number of cycles.

The importance of the proper amount of tip relief was emphasized by the results of the Lot 2 tests. The initial tests in this series were conducted at the 1500-pound load level with gears whose tip relief was in the order of .0020 inch. Three of these gears failed in less than 500,000 cycles. When the tip relief was reduced to approximately .0017 inch, the failure times at this load were more reasonable as shown in Figures 12 and 15.

Tip relief is incorporated in the test gear to compensate for gear tooth deflection under load. Obviously, the higher the load for a given gear, the more deflection and the more tip relief required; thus, a gear can be designed to run at a given load level by modifying the profile to compensate for tip deflection at that particular load. When the gear is run at loads substantially higher than the design load, tip scoring results. When it is run at lower loads, load concentrations at the pitch line result, which could cause premature failure by pitch line pitting or mid-tooth fracture.

A practical solution to this problem evolved from the Heat 2 tests. A lower value of tip relief was used for the two lower load levels and a higher value for the two higher load levels.

The average fatigue strength at 10^8 cycles for all of the processes tested is 1460 pounds. The AGMA tooth bending stress at this load level is 96850 psi when a load distribution factor of 1.3 is used to account for end-loading. The corresponding average stress in the single-tooth fatigue program of Reference 2 was 120,500 psi. The ratio of these stresses is a measure of the dynamic effect at 9200 rpm. Thus,

$$K_v = \frac{96850}{120500} = .80$$

where K_v is the experimental dynamic factor. This value agrees with the results of the dynamic tests of Reference 7 within 10 percent.

METALLURGICAL STUDY

INVESTIGATION PROCEDURE

Metallurgical investigations were conducted on the failed gears of each heat to determine the mode of failure, origin of failure, microstructure of case and core, and hardness of the case and core.

Fourteen failed test gears were subjected to a material analysis and mode of failure examination as follows:

1. The fractured teeth were examined under a low-power stereo-microscope to determine the origin and mode of failure.
2. One fractured tooth from each gear was further examined to determine R_c hardness values for the case and the core.
3. The fractured teeth were mounted, polished, and etched with a 2 percent Nital solution to determine the microstructure of the case and core.
4. The effective case depth was determined by a visual examination of the etched mounts under a Brinell microscope. The case-core interface was taken at R_c 50.

RESULTS

All 14 gears were found to have at least one tooth that failed by bending fatigue in the root fillet area. These fatigue failures were accompanied, in some cases, by static fracture and cracking in adjacent teeth. Individual tooth fractures are shown in Figures 18 through 21. The "F", "S", and "C" denote fatigue, static, and cracked respectively.

Two modes of cracking were evident: multiple fatigue cracks across the tooth face which had propagated to fracture, and single cracks (the predominant mode) at the junction of the root radius and gear tooth profile. These modes of failure are illustrated in Figure 22.

The gear material of each heat met the chemical requirements of AMS 6265 (9310 CVM steel). Microstructural observations and hardnesses are shown in Table XIII. As indicated, the effective case depth of the gears of Heat 1 selected for analysis is below the specified minimum depth requirement. This is because all of the gears in this heat were reground to increase the tip relief after initial testing showed signs of excessive tip loading. This condition would not affect the comparative test results.

Metallographic examination revealed a typical martensitic microstructure although some degree of carbide network and retained austenite was observed in the test samples from Heat 1. Inspection of the test data,

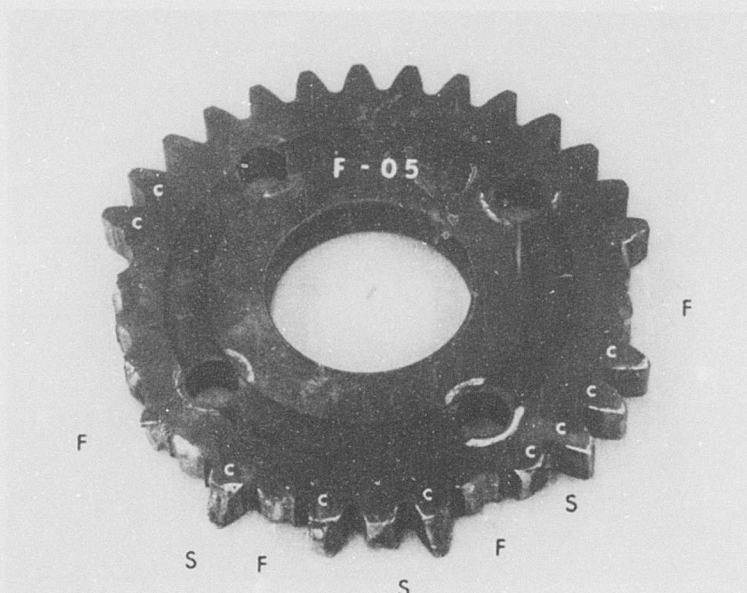
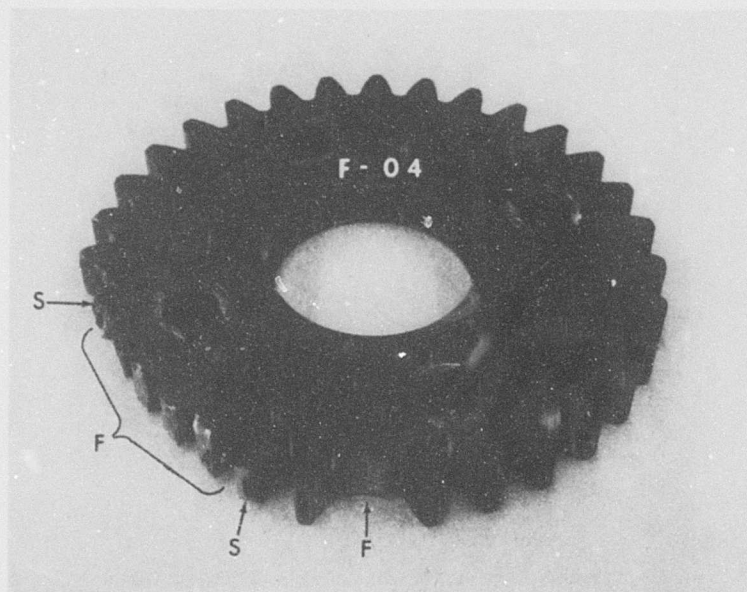


Figure 18. Failed Test Gears - Process "A".

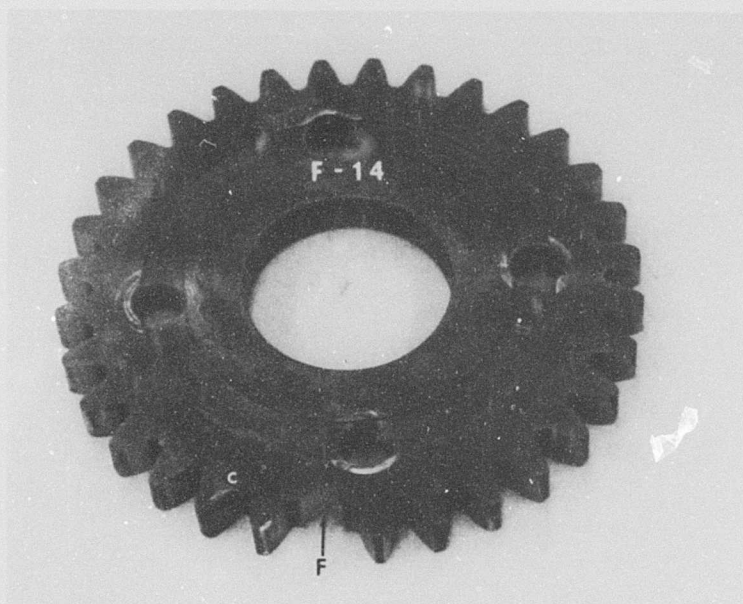
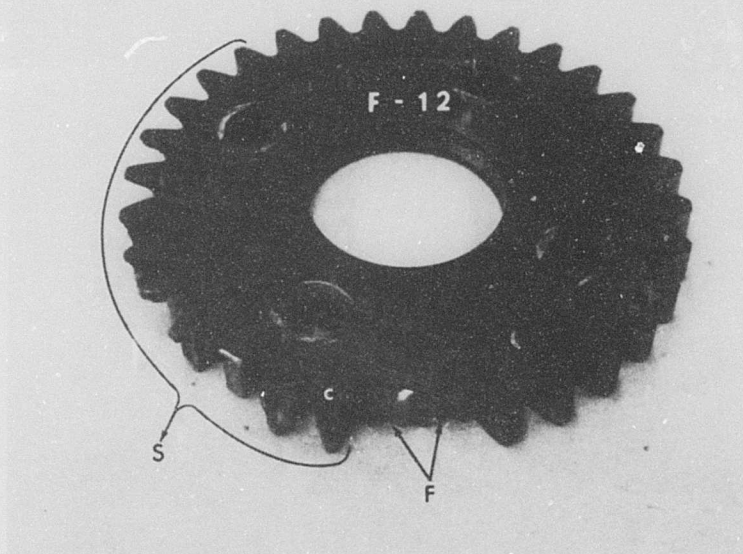


Figure 19. Failed Test Gears - Process "B".

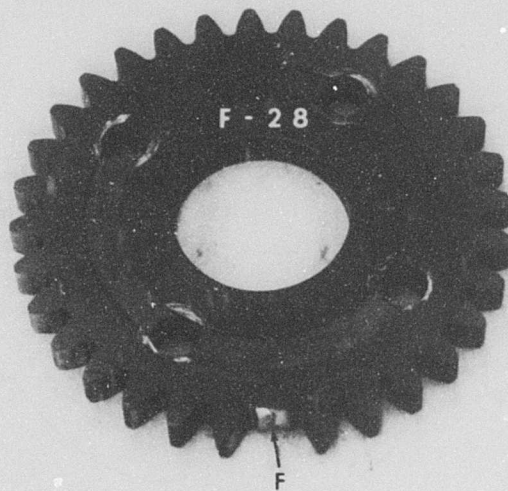
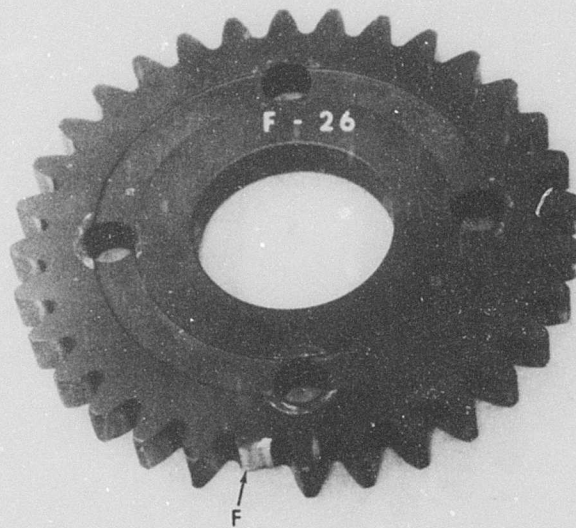


Figure 20. Failed Test Gears - Process "C".

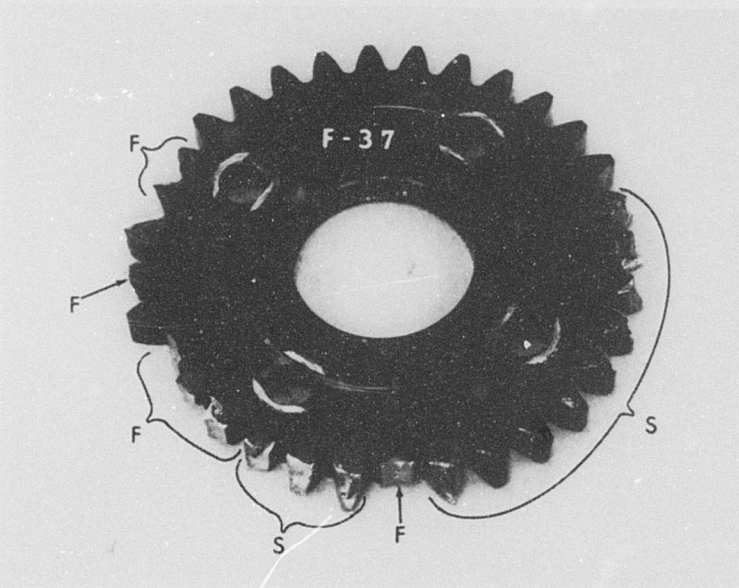
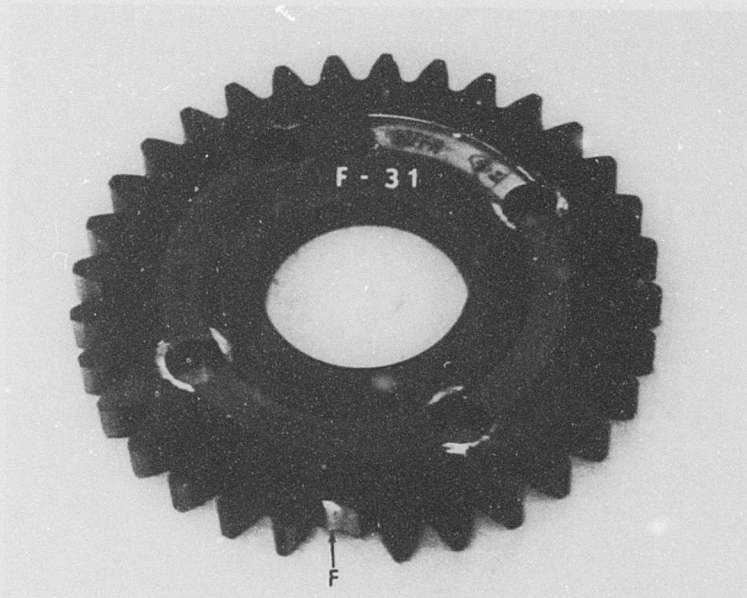
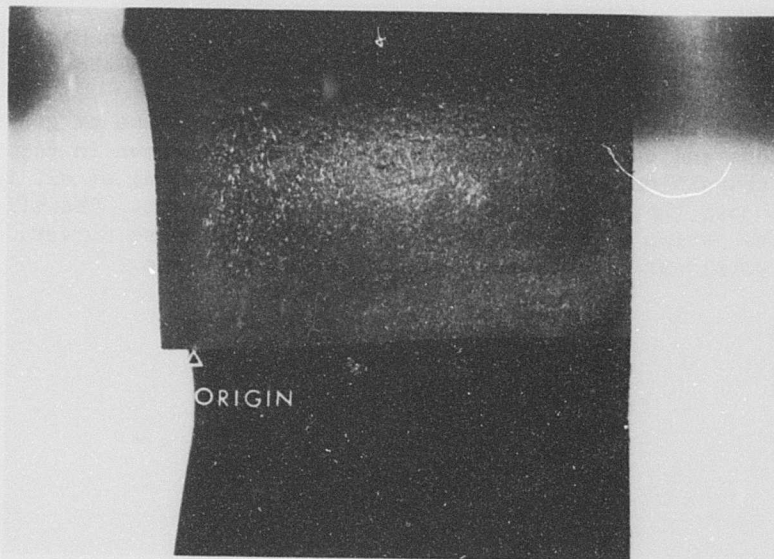
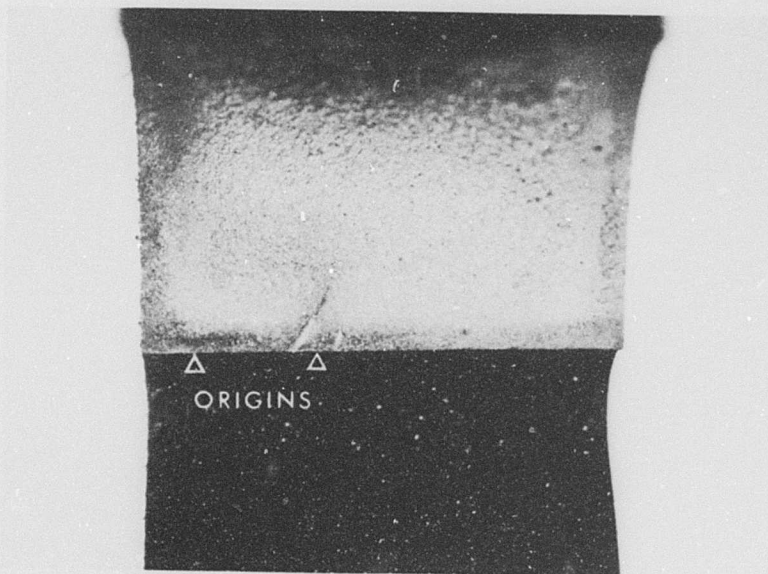


Figure 21. Failed Test Gears - Pancake Forgings.



(a) Fracture Interface
Gear S/N F28



(b) Fracture Interface
Gear S/N F30

Figure 22. Modes of Gear Fracture.

however, reveals that the test gears with the higher percentage of retained austenite did not consistently fail earlier than the others.

Microhardness traverses were taken across the pitch line of gears from each process and each heat. Typical traverses are shown in Figure 23. A smooth transition from case to core, which is typical of all the gears checked, shows that a good carburized case was present. The effective case depths, which were obtained from the microhardness survey, agreed with the total case depths determined by optical means.

TABLE XIII. SUMMARY OF METALLURGICAL
OBSERVATIONS OF TEST GEARS

Heat No.	S/N	Case Hardness (R _c)	Core Hardness (R _c)	Effective Case Depth (in.)
1	F04	62	36	.028
	F05	60	34	.032
	F12	60	34	.028
	F14	62	35	.030
	F26	62	36	.032
	F28	58	37	.028
	F31	60	40	.028
	F37	60	39	.028
2	F05	60	38	.040
	F28	60-61	37	.038
	F30	61-61.5	37	.040
	F51	61.5	38	.040
	F53	61-61.5	38	.035
	F69	61	38	.040
Required		58-64	34-40	.035-.043

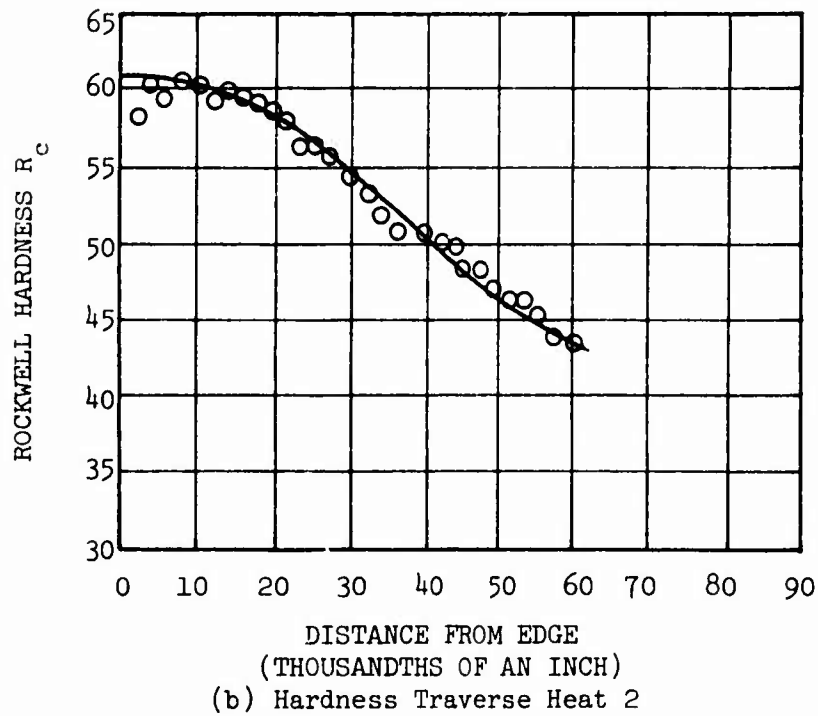
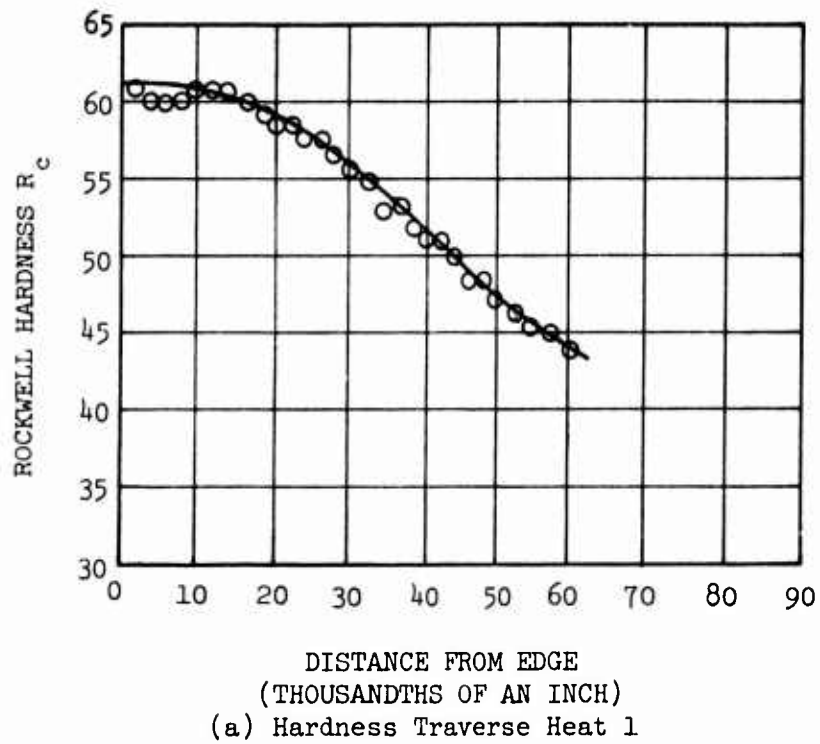


Figure 23. Typical Microhardness Readings.

CONCLUSIONS

1. On the basis of the dynamic fatigue tests conducted on gears produced by three high-energy gear forging processes and a conventional forging process, the best advanced forging process shows an advantage, with respect to gear tooth bending strength at 10^8 cycles, of 5 to 8 percent over the conventional process. The advanced forging process which was ranked lowest had a fatigue strength which was 98 percent of the conventional process.
2. The results of additional testing conducted on a second heat of the same material were almost identical to the results of Heat 1 with regard to percent improvement over the conventional process.
3. Quantitative evaluation of the three parameters of the standard S-N relation used in this program can be difficult if a limited number of test points are used or if the points are grouped in clusters, thus decreasing their effectiveness in defining the S-N curve shape.
4. There is no appreciable difference in the fatigue strength of high-energy gears processed without the green-grinding operation and those which were green-ground prior to hardening and final grinding. Elimination of the green-grinding, besides simplifying the processing, means that less of the beneficial grain flow on the tooth surface produced by the forging operation will be removed during grinding. To take advantage of this, a smaller material allowance will have to be designed into future forging dies.
5. The dynamic effect at 9200 rpm is shown to account for a significant reduction in fatigue strengths for both the advanced forging processes and the conventional forging. In this program the average dynamic fatigue strength of the gears tested is 80 percent of the fatigue strength of gears tested in the non-rotating or single-tooth mode. This result is consistent with design factors used in high-quality gearing.
6. The magnitude of the tip relief incorporated in highly loaded gears to compensate for tooth deflections can have a significant effect on the test results. Too high a value can severely limit the gear fatigue life. In this program a .0003 decrease in tip relief (from .0020 inch to .0017 inch) resulted in an increase in life, at the 1500-pound load level, from 500,000 cycles to approximately 30 million cycles.

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